SCHALLER

The Theory and Design of a Turbo-Blower

Mechanical Engineering

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THE THEORY AND DESIGN OF A TURBO-BLOWER

BY

ALWIN LOUIS SCHALLER

B. S. UNIVERSITY OF ILLINOIS, 1907

THESIS

SUBMITTED IN PARTIAL FULFILLMENT

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I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

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THE THEORY AND DESIGN OF A TURBO VOLUME BLOWER.

I - INTRODUCTION.

The design of centrifugal fans in the past has largely been carried out with the aid of emperical equations, the constants for which were determined by a series of experiments. Such a method of procedure has not been conduaive to the development of a machine of high mechanical efficiency nor has it brought out the best forms of operating characteristics. The bulletins of the leading makers contain but few references to efficiency and curves showing the pressure and horse-power characteristics are entirely absent; as a natural result many blowers are performing their work with an entirely unnecessary expenditure of mechanical energy. This is not due to a lack of knowledge of the fundamental laws which govern the design and performance of centrifugal fans, but rather to a lack of their scientific application.

The field which has offered the widest use of the blower is that of ventilation. In this service the pressures required are comparatively small, so that the power necessary does not assume very large proportions. For this reason, perhaps, but little attention has been paid to the mechanical efficiencies of this type of machine. The recent introduction of high pressure mechanical draft, requiring in addition, large volumes of air, has created a demand for an efficient blower of a good mechanical construction.

It is the object of this thesis to present the operating conditions which are encountered in the applidation of the centrifugal blower to high pressure forced draft service and

to indicate the method of determing the data necessary for carrying out the complete design of such a machine. This information is preliminary to a presentation of the underlying theory of centrifugal fans. In order to prove that the theory is borne out by the results of tests an analytical investigation of published blower trials is included. The general construction is worked out in such a way that the blower will be particularly suited for steam turbine drive. In addition to the complete design, the operating characteristics are approximately predicted and a method for testing this type of blower is briefly outlined.

Acknowledgement is due to Mr. C. V. Kerr for many valuable suggestions, and for some of the mechanical details incorporated in the general design. Mr. J. L. Moore, Chief Engineer of the Kerr Turbine Company kindly supplied the data for small steam turbines which was necessary to make this thesis complete.

II - THE REQUIREMENTS OF A HIGH PRESSURE FORCED DRAFT INSTALLATION.

In an electric power plant the substitution of turbines for reciprocating engines resulted in a larger demand for steam on account of the increased output of the station. Lack of sufficient room prevented the installation of the desired boiler capacity. However, by the addition of a battery of two 500 H. P. boilers the demand for steam could be supplied, provided, that the new boilers could be operated efficiently and continuously at 100 percent above their normal rating.

Recent boiler trials conducted by D. S. Jacobus at the Detroit Edison Company have shown that it is easily possible with proper furnace conditions to attain a capacity far in excess of builders rating with practically constant efficiency. While a combined efficiency of boiler and grate of 80 percent was obtained at normal rating, this efficiency was only reduced to 76 percent while the capacity of the boiler was doubled.

These remarkable results were obtained with Taylor Stokers. In this stoker the ability to burn efficiently the maximum amount of coal per hour is made possible by the use of a large number of retorts per boiler. Air is supplied to these retorts from a wind box connected to the main duct through a tuyer.
In order to obtain complete and smokeless combustion it is essential that the fuel and air come into intimate contact at the kindling temperature and in proper proportions. In this system of stoking the first condition is obtained by operating with a deep fuel bed and introducing the raw coal at the bottom. However, a deep fuel bed offers so much resistance to the flow of air that mechanical draft becomes necessary. In this way a ready means is

provided for regulating the air supply to correspond with the varying rate of combustion. This fulfills the second requirement for high furnace efficiency.

By the analysis of continuous samples of flue gas the proper ratio between air and coal supply is determined. The relative speeds of the stoker and blower are then fixed and perfect regulation is always secured. Such a determination of the amounts of air and coal insures the highest possible efficiency throughout a wide range of load. It becomes impossible to introduce more coal than can be properly burned nor can more air be admitted to the furnace than is necessary to support combustion.

From these considerations it is plainly evident that the Taylor Stoker is particularly fitted for the service required under the conditions obtaining in the power plant under discussion. Assuming that this method of stoking is adopted our problem consists in the determination of the proper capacity of the blower, and the air pressure required for supplying the necessary mechanical draft, after selecting the best form of driving power to carry through the complete design of the blower and to outline the method of operation for securing the best results under the imposed conditions.

The component parts of the general problem will be taken up in the order named. The boilers which are to be supplied with mechanical draft have a nominal aggregate capacity of 1,000 Boiler Horse Power. With a fair grade of coal an equivalent evaporation of 8.5 lbs. of water per pound of coal may safely be assumed.

Therefore

$$\frac{34.5}{8.5} = 4.05 \text{ lbs. of coal per hour per B. H. P.}$$

Taking 18 lbs. of air to burn one pound of coal we have

$$18 \times 4.05 = 73 \text{ lbs. of air per hour per B. H. P.}$$

$\frac{73 \times 13.8}{60} =$ 16.8 cubic feet per minute per boiler H. P.

The total demand upon the blower for various capacities will then be

1000 B. H. P.	Normal Rating	16,800	cubic feet per minute.
1500 "	150% of "	25,200	" " " "
2000 "	200% of "	33,600	" " " "

As these boilers will be required to operate at maximum capacity a large share of the time the blower should be designed large enough to supply the maximum demand efficiently. The delivery of a given blower can be increased by an increase of speed. However, the maximum efficiency will occur at a higher pressure. If this battery of boilers will be capable of developing more than the capacity given above there will of course be a larger demand upon the blower. This increased delivery can be taken care of by a variation of speed. Within reasonable limits the economical operation of the unit will not be impaired because a greater delivery will require a higher pressure. For this reason if the normal capacity of the blower is fixed at 30,000 cubic feet per minute at the proper pressure, the aditional demand for air can easily be taken care of.

We are now confronted with the problem of determing the compression required at normal rating and the variations of pressure necessary for the various rates of firing. In addition we must devise some means of regulation so that the quantity of air supplied will be equal to that required by the process of combustion.

It will be assumed here that the thickness of fuel bed will be independant of the rate of stoking. For lower capacities the fuel will simply be introduced in smaller quantities. The same being true of the air supply. This results in the development of

less boiler horse power. With this condition of operation the resistance of the fuel bed to the passage of air will be constant. In other words this resistance will constitute some fixed orifice for the fan. With a fixed orifice the quantities will vary directly as the velocities. Now the velocities vary directly as the square roots of the pressures, therefore, the delivery of air will vary directly as the square root of the pressure. By means of this relation when the necessary head for a given delivery is found the head for any other delivery can readily be computed.

For normal capacity of the stokers which in this case corresponds very closely to 2000 boiler horse power the makers call for a static pressure of 5 inches of water at the tuyers. These tuyers, two in number have an area of 3.5 square feet each. The air velocity through the tuyers is therefore,

$$\frac{30,000}{60 \times 2 \times 3.5} = 71.5 \text{ ft./ sec.}$$

The total pressure required at blower is equal to the static head plus the velocity head at the tuyers plus the losses of head in the air duct between the blower and the stokers.

In installations not originally designed for mechanical draft considerably difficulty is usually encountered to find room for the blower unit as well as the air duct. To reduce the required space to a minimum we will employ the same velocity in the duct as in the tuyers. While this is considerably larger than good practice prescribes and results in larger friction losses, it reduces the required space. Taking the dimensions as 2'-0" x 3'-6" we have the cross section equal to the total area of tuyer openings. From the layout of the boiler plant the length of the duct between blower and stokers is found to be 50 feet. In this length there are three elbows.

From W. D. Taylors experiments the loss due to the pressure of three 90° elbows is equivalent to the addition of 10 feet to the length of the main. For a rectangular main of short side h and long side nh, he found the loss of head to be

$$H = \frac{1+n}{n} \times \frac{e}{h} \frac{v^2}{22,500,000}$$

e = length of pipe in feet.
v = velocity in feet per minute.
H = loss of head in feet of air.

In this case

e = 60 feet
h = 2 feet
n = 1.75
v = 4300 feet per minute.

Substituting

$$H = \frac{2.75}{1.75} \times \frac{60}{2} \times \frac{4300^2}{22,500,000}$$

$$= 38.8$$

For average conditions 68.3 feet of air = 1 inch of water. Therefore, loss of head = .6 inch of water. To this loss of head we must add the loss incurred at the branches of the duct and the losses in the connections to the tuyers . For the total loss of head one inch of water may be taken as a conservative estimate. Therefore if the blower delivers the air at duct velocity a static pressure of six inches of water will be required at entrance to the duct to maintain a static pressure of five inches of water in the tuyers . Under these conditions nothing can be gained by discharging the air from the blower casing at a lower velocity than that in the duct. With a large discharge velocity the volute becomes smaller and a saving is thus effected in both weight and overall dimensions of the unit. The assumed pipe velocity corresponds to a head of 1.1 inches of water. Consequently the blower will be required to deliver 30,000 cubic feet of air per minute

against a total pressure of 7.1 inches of water measured at the discharge opening in order to maintain a static pressure of five inches of water at the tuyers . This total pressure is equal to the sum of the static and velocity head plus the loss of head in the air duct.

When the battery is developing 1500 H. P. about 25000 cubic feet of air will be required. To pass this quantity of air through the fuel bed the necessary static pressure is

$$30{,}000 : 25{,}000 :: \sqrt{5} : \sqrt{x}$$

$$x = \frac{25^2 \times 5}{30^2}$$

$= 3.5$ inches of water.

The duct velocity is $\frac{25}{30} \times 71.5 = 59.5$ ft./sec.

The loss in the duct now is

$$H = \frac{2.75}{1.75} \times \frac{60}{2} \times \frac{\overline{59.5 \times 60}^2}{22{,}500{,}000}$$

$= 26.8$ feet of air,

$= .4"$ of water.

The head corresponding to the duct velocity of 59.5 ft./sec. is equal to

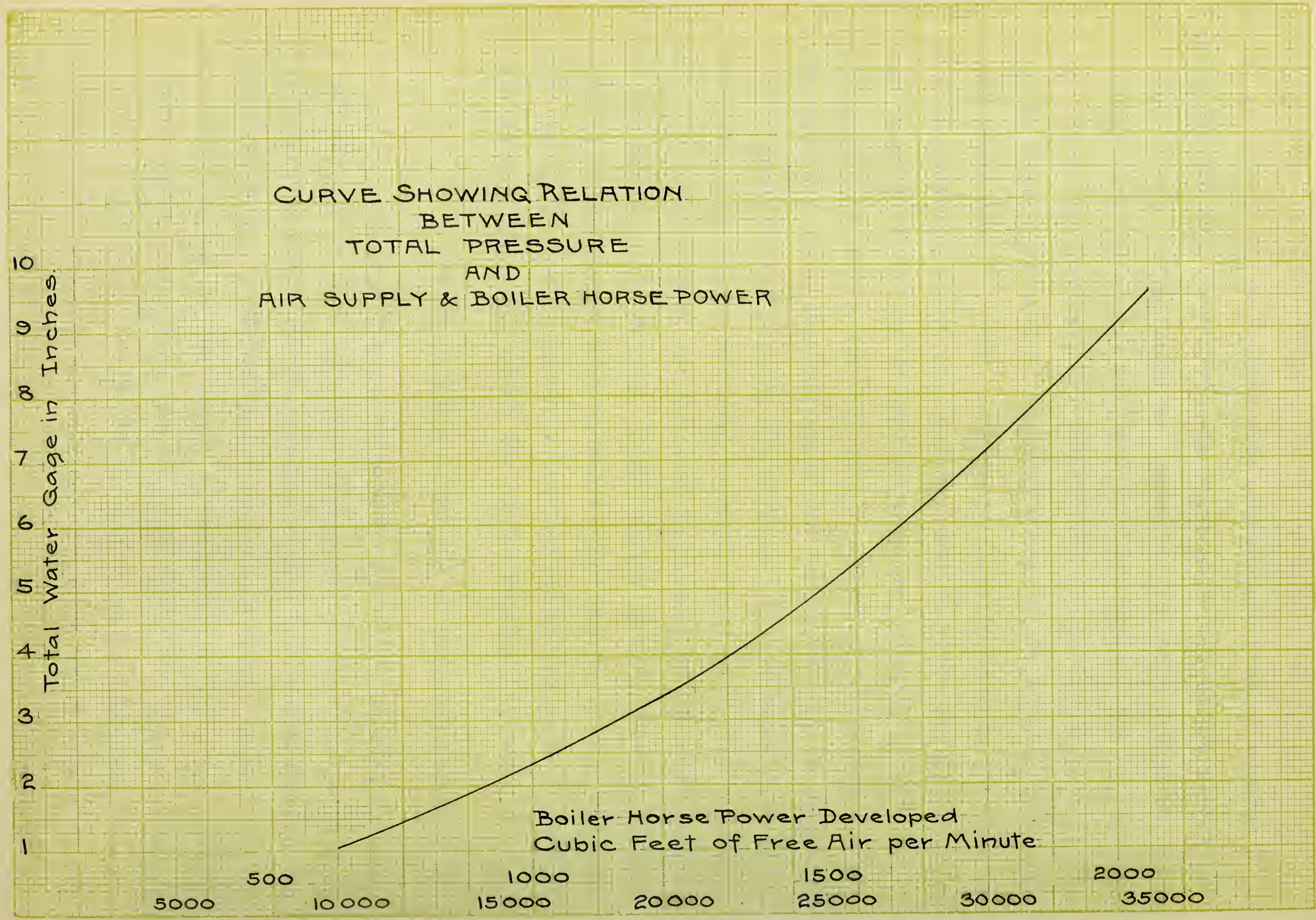
$$H = \frac{59.5^2}{2 \times 32.2}$$

$= 55$ feet of air

$= .8$ inch of water.

Allowing .2 of an inch for losses unaccounted for the total pressure at the blower for a delivery of 25000 cubic feet of air will be 4.9 inches of water.

In the same way the pressures required for the entire range of delivery of the fan were computed and the results plotted in the form of a curve drawn on page 9 . With the aid of this curve the total pressure required at the discharge opening of the blower for any given delivery or corresponding boiler horsepower can at once


CURVE SHOWING RELATION
BETWEEN
TOTAL PRESSURE
AND
AIR SUPPLY & BOILER HORSE POWER
Total Water Gage in Inches.
1
2
3
4
5
6
7
8
9
10
Boiler Horse Power Developed
Cubic Feet of Free Air per Minute
500
1000
1500
2000
5000
10000
15000
20000
25000
30000
35000
EUGENE DIETZGEN CO., CHICAGO.

be determined. As the rate of firing is varied the speed of the fan will have to be altered in such a way that the total head will always correspond to that upon the diagram. It is now clearly evident that if the conditions for high boiler and furnace efficiency previously outlined are to obtain the blower supplying air to the stokers cannot operate at any fixed speed. This unit must be so constructed that a wide range of speed variation can readily be secured and easily controlled. For this form of service the steam turbine is admirably adapted as a driving motor.

III - CHARACTERISTICS OF THE SMALL STEAM TURBINE.

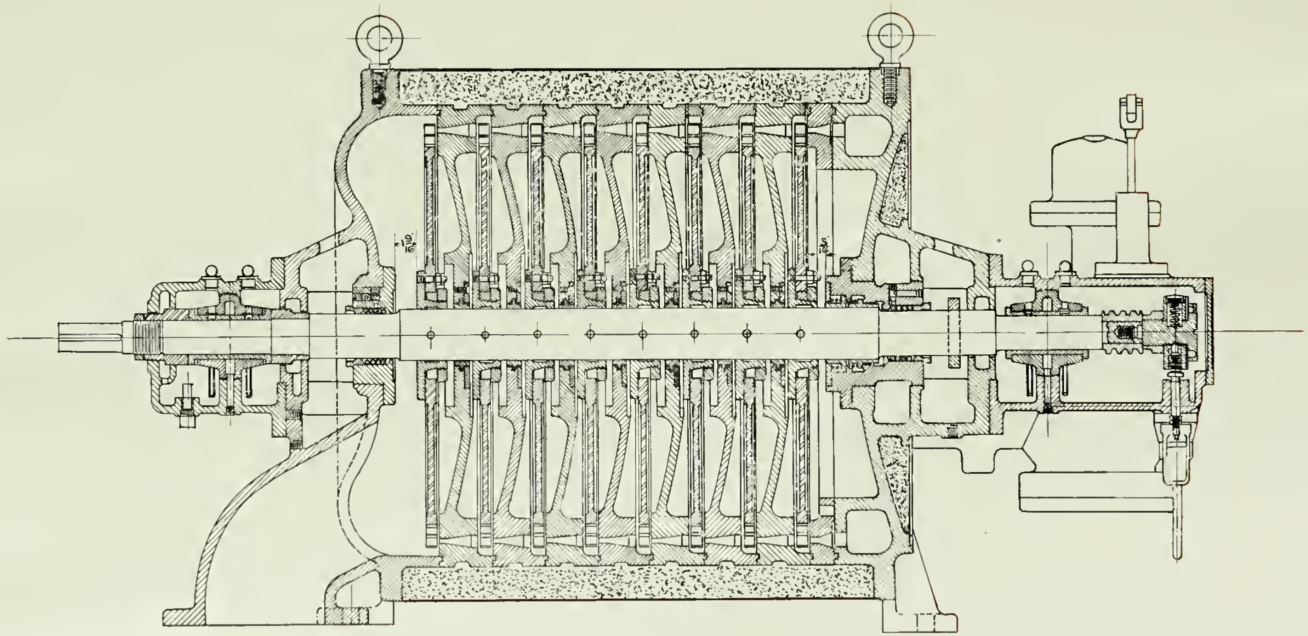
In selecting driving motors for power plant auxiliaries the prime consideration is reliability followed closely by plant economy. In this field the small steam turbine is unexcelled by any other available form of motive power. Within a wide range any rotative speed can be secured by varrying the pressure upon the first stage nozzles. Provided of course that the shaft horse power remains within the limits for which the turbine was designed. Furthermore, it is constructed and installed similar to the main units and can therefore, be operated with the same degree of reliability.

It has sometimes been assumed that by employing electric drive the auxiliaries could be operated with the same efficiency as the main unit. This is entirely erroneous. When the main units are operating condensing the steam from all of the auxiliaries is required for heating boiler feed water. In this way nearly all of the heat in the exhaust steam is returned to the boilers instead of being thrown into the condenser, as would be the case if the power were indirectly supplied by the main unit. The exhaust steam of a turbine is well adapted for feed water heating because it is

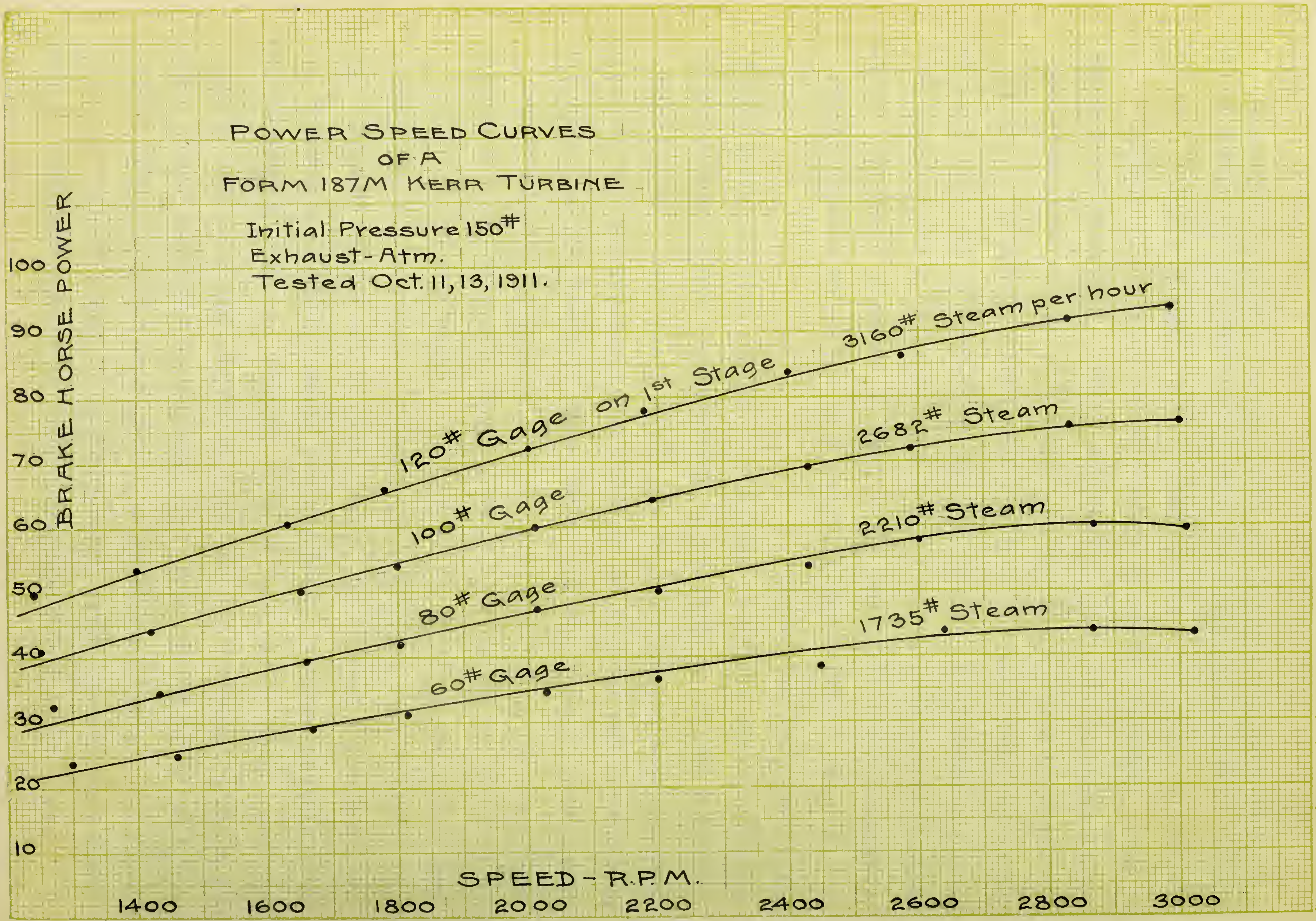
free from oil. Such complications as oil separators are therefore, entirely eliminated.

It sometimes happens that more exhaust steam is available than is required for heating boiler feed water. In such a case the turbine can be operated condensing. The progress made in the development of the small turbine has decreased the steam consumption when operating condensing to such a point that no other form of driving power either steam engine or electric motor could be substituted and increase the plant economy.

On page 12 is shown a sectional view of an eight stage turbine built by the Kerr Turbine Company. The characteristics of this particular type of turbine are well brought out by the power speed curves on page 13 . These curves,which are often termed the calibration curves, show the relation between speed and horse power output at constant steam pressure. For the first half of the range the horse power is very approximately a direct function of the rotative speed. The out put of the machine increases as the peripheral velocity of the buckets approaches its economical value. The economical velocity being smaller for the lower ratios of expansion the maximum output for lower steam pressures is obtained at a slower speed than for the higher pressures. For example, the curve for 60 lbs. pressure reaches a maximum at 2900 revolutions per minute, while the curve for 120 lbs. pressure has no maximum within the limits of the diagram. The great difference in power out put at different speeds is more forcibly brought out by actual values. With a steam pressure of 120 lbs. per square inch upon the first stage nozzles the brake horse power at 1200 revolutions per minute was 47. When the speed was increased to 3000 R. P. M. it was 93. In other words the output increased 100 percent.



Sectional view of Type M Economy Turbine, Kerr Turbine Company, Wellsville, N. Y.



POWER SPEED CURVES
OF A
FORM 187M KERR TURBINE
Initial Pressure 150#
Exhaust-Atm.
Tested Oct. 11, 13, 1911.
BRAKE HORSE POWER
10
20
30
40
50
60
70
80
90
100
120# Gage on 1st Stage
3160# Steam per hour
100# Gage
2682# Steam
80# Gage
2210# Steam
60# Gage
1735# Steam
SPEED - R.P.M.
1400
1600
1800
2000
2200
2400
2600
2800
3000

The calibration curves also show the relation between initial steam pressure and brake horse power at constant speed. For any given speed equal increments of steam pressure give equal increments of brake horse power. In other words the horse power out put is a linear function of the steam pressure upon the first stage nozzles. When, therefore, a steam turbine has been calibrated and the relation between horsepower and steam pressure determined, it can be used to measure the power input to the driven machine. If, for example, this turbine were directly coupled to a centrifugal pump operating at 2,000 R. P. M. with a steam pressure upon the first stage of 100 lbs. per square inch; reference to the diagram shows that under these conditions the turbine is developing 59 brake horse power. For any other steam pressure the corresponding horse power can readily be obtained by interpolation. In operating steam turbine driven machines the brake horse power absorbed can therefore easily be obtained by observing the steam pressure upon the first stage nozzles. Provided of course that the turbine has been previously calibrated.

Another striking characteristic of the steam turbine is the relation between speed and steam flow at constant pressure. If the steam pressure were held constant and the total steam consumption per hour determined for various speeds throughout the entire range the values thus obtained would be very nearly constant. That is the steam flow for a turbine is independant of the rotative speed and is fixed by the initial pressure alone. In commercial testing of small steam turbines the steam flow is obtained by clamping the turbine shaft in a fixed position and then maintaining constant initial pressures and weighing the condensate. The constant pressure lines upon the calibration curve sheet therefore become

constant steam flow lines and indicate the steam economy of the turbine at various speeds. These curves illustrate very forcibly the relation between steam consumption and rotative speed. With an initial steam pressure of 120 lbs. per square inch the total steam per hour is 3160 pounds. At 1200 r. p. m. when the turbine is developing 47 brake horse power the steam consumption is 67.3 lbs. of steam per horse power hour. At 3000 r. p. m. when the turbine is developing 93 brake horse power the steam consumption has fallen to 34 lbs. of steam per horse power hour.

The calibration curves exhibit still another interesting relation. When the steam flow for various steam pressures has been determined and indicated as shown on these curves the relation between steam economy and steam pressure at constant speed can be deduced . A series of auxiliary curves can be drawn which will show the variation of steam consumption per horse power hour with initial steam pressure for any given rotative speed.

In considering the economy of a turbine driven unit, such as a turbo-pump or a turbo-blower the combined efficiencies of both machines must be taken into account. The steam consumption per water or air horsepower now becomes the essential factor. In designing directly connected machines for turbine drive the efficiencies of both machines should be so chosen that their product is a maximum. In the steam turbine the economy is clearly a function of the rotative speed. Considering the turbine alone the lowest steam consumption will be obtained at its most economical speed. This speed, however, usually lies above the possible range for the driven machine. When it is impossible to operate at the most economical speed of the turbine the greatest overall efficiency can usually be obtained by making the speed of the driven machine as high as possible even if

some sacrifice of mechanical efficiency must be made. The gain on the turbine will more than offset the loss thus incurred. This can be illustrated by a concrete example. Let the efficiency of a blower operating at 1200 r. p. m. be 67%. The water rate of the turbine at this speed is 67.3 pounds per horse power hour.

$$\frac{67.3}{.67} = 104.4 \text{ lbs. of steam per air horse power hour.}$$

Assume that by raising the speed to 2000 r. p. m. the mechanical efficiency were lowered to 56 percent. The water rate of the turbine per horse power hour at this speed is 41.1 pounds.

$$\frac{41.1}{.56} = 75.5 \text{ lbs. of steam per air horse power hour.}$$

In this case the increase of speed has reduced the steam consumption from 104.4 to 75.5 pounds of steam per air horse power hour even though the blower efficiency was lowered from 67 to 56 percent.

IV - THE THEORY OF THE CENTRIFUGAL BLOWER.

The following presentation of the theory of centrifugal fans is due to Weisbach.

In all fan blowers the pressure of the air is changed by a change of its state of motion. This may be accomplished by a change of direction or a change of velocity, or by a combination of both. In fans of the propeller wheel type, the change of pressure is principally due to a change of velocity. In the centrifugal blower, the increase in pressure is caused by the change in the direction of motion, or the resulting centrifugal force.

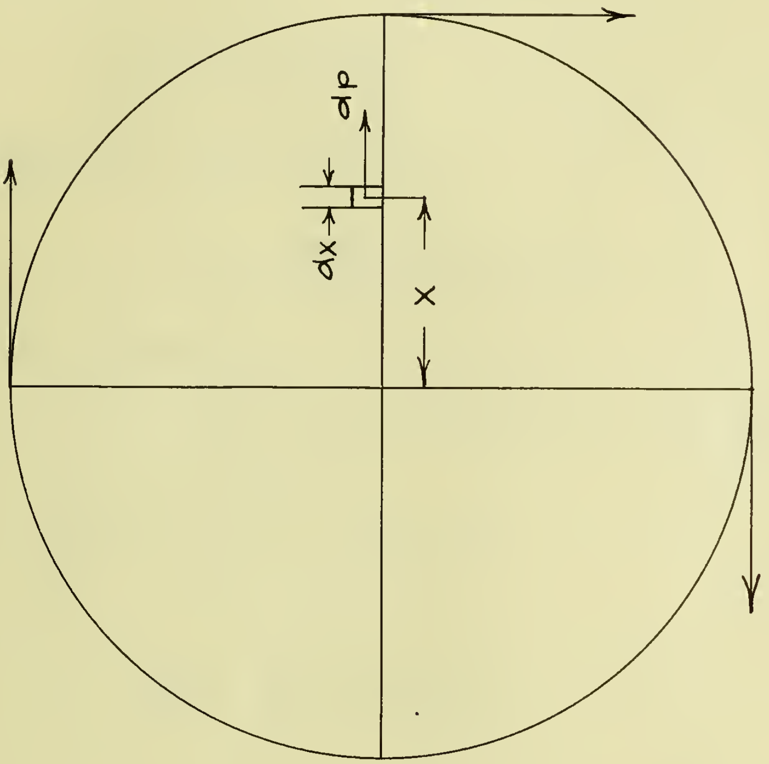


Figure 1.

Let us first assume that the radial velocity of the air in a fan is small in comparison with the velocity of rotation, and that the vanes are straight.

Let w = angular velocity of the impeller shown in figure. The velocity of any point distance x from the center is

$$u = x\omega$$

At this radius, consider an element of air having unit cross-section, a thickness d x, and the specific weight y. The centrifugal force set up by this element will be

$$dp = x\omega^2 \frac{dx\,\gamma}{g}$$

According to Mariottes law

$$\gamma = \frac{.0055\ p}{1+.00203(t-32)}\ \text{lbs.}$$

Let $\gamma = \phi p$ where p is the pressure in pounds per square inch, and ϕ is a coefficient depending on the temperature t, which is usually constant.

Substituting for γ, we have

$$\frac{dp}{p} = \frac{\phi}{g}\omega^2 x\,dx.$$

Now if r_1 represents the inner radius of the wheel, and r_2 the outer radius, and p_1 and p_2 represent the pressure at the inner and outer circumferences respectively, we have by integration:

$$\int_{p_1}^{p_2} \frac{dp}{p} = \frac{\phi}{g} \omega^2 \int_{r_1}^{r_2} x dx$$

$$\log e \frac{p_2}{p_1} = \frac{\phi}{g} \omega^2 \left(\frac{r_2^2 - r_1^2}{2} \right)$$

But as $r_1 \omega = u_1$ and $r_2 \omega = u_2$

$$\log e \frac{p_2}{p_1} = \frac{\phi}{g} \left[\frac{u_2^2 - u_1^2}{2} \right] \quad \text{or} \quad p_2 = p_1 e^{\frac{\phi}{g}\left[\frac{u_2^2 - u_1^2}{2}\right]}$$

As the exponent $\frac{\phi}{2g}[u_2^2 - u_1^2]$ is always small, we may place

$$e^{\frac{\phi}{2g}[u_2^2 - u_1^2]} = 1 + \frac{\phi}{2g}[u_2^2 - u_1^2] = \frac{p_2}{p_1}$$

Then by subtracting 1 from both sides

$$p_2 - p_1 = \phi p_1 \left[\frac{u_2^2 - u_1^2}{2g} \right] = y_1 \left[\frac{u_2^2 - u_1^2}{2g} \right]$$

If instead of the pressure p_1 and p_2 we use the corresponding heights of the water barometer b_1 and b_2, we have

$$b_2 - b_1 = \phi b_1 \left[\frac{u_2^2 - u_1^2}{2g} \right] = \frac{y_1}{y_0} \left[\frac{u_2^2 - u_1^2}{2g} \right] = \frac{1}{\varepsilon_1} \left[\frac{u_2^2 - u_1^2}{2g} \right]$$

where $\varepsilon_1 = \frac{y_0}{y_1}$ is the ratio of the density of water to that of the entering air. For the case in which the outer radius is very much greater than the inner, we have approximately

$$b_2 - b_1 = \frac{u_2^2}{2g\varepsilon}$$

If the air flows directly into the atmosphere from the wheel, as in some exhausters, the velocity of efflux, which is nearly equal to the circumferential velocity, becomes zero without exerting any action. Here b_2 is equal to b_0, the barometric height of the atmosphere. The manometer height in the supply pipe will now be negative and equal to

$$h = b_0 - b_1 = \frac{u_2^2}{2g\varepsilon_1}$$

In the corresponding case of a blower without diffuser, we have

$$h = b_2 - b_1 = b_2 - b_0 = \frac{u_2^2}{2g\varepsilon_1}$$

If we surround the fan wheel with a casing so that the velocity u_2 is gradually reduced to zero, its kinetic energy will be trans-

formed into pressure measured by a water column of a height

$$\frac{u_2^2}{2g\varepsilon_1}$$

Let b_2' be the outer pressure with a fan with casing. For an exhauster

$$b_0 = b_2' = b_2 + \frac{u_2^2}{2g\varepsilon_1} = \left(b_1 + \frac{u_2^2}{2g\varepsilon_1}\right) + \frac{u_2^2}{2g\varepsilon_1} = b_1 + \frac{u_2^2}{g\varepsilon_1}$$

Let H be the manometer height; then

$$H = b_2' - b_1 = b_0 - b_1 = \frac{u_2^2}{g\varepsilon_1} = 2h$$

For a blower

$$b_2' = b_2 + \frac{u_2^2}{2g\varepsilon_1} = \left(b_1 + \frac{u_2^2}{2g\varepsilon_1}\right) + \frac{u_2^2}{2g\varepsilon_1} = b_0 + \frac{u_2^2}{g\varepsilon_1}$$

from which

$$H = b_2' - b_0 = \frac{u_2^2}{g\varepsilon_1} = 2h$$

In both cases the manometer height has been doubled by the use of a casing. On account of the friction of the air in the casing and other disturbing influences, the manometer height is considerably smaller than $\frac{u_2^2}{g\varepsilon_1}$

The work theoretically required is

$$W = Qh\gamma_0 = Q\frac{u_2^2}{g\varepsilon_1}\gamma_0 = Q\frac{u_2^2}{g}\gamma_0$$

where Q is the quantity of air delivered per unit of time. The work actually required is considerably greater, and is

$$W = \frac{Qh\gamma_0}{\eta}$$

where η is the efficiency of the machine.

In most cases, however, the conditions assumed in the preceeding discussion do not obtain. The relative velocities are too large to be neglected, and the vanes of blower impellers are often curved instead of radial.

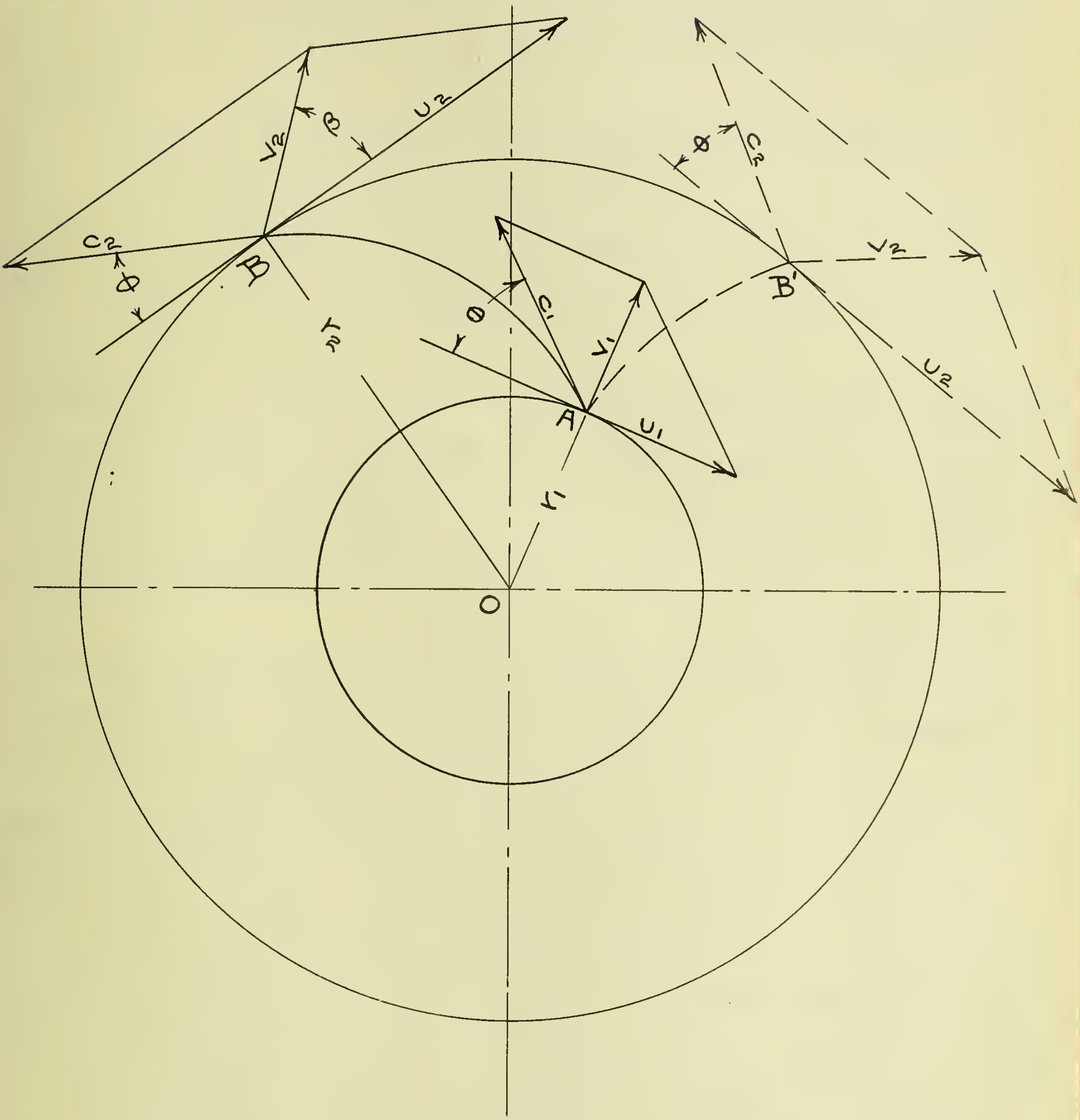


FIGURE 2.

In the impeller shown in the figure 2 AB is one of a number of curved vanes. The air enters the wheel along the axis O in such a way that it is evenly deflected in all directions. In other words, the relative velocity of entrance to the vanes is radial.

Let u_1 = peripheral velocity at inner rim of wheel

v_1 = absolute velocity of entrance

c_1 = relative velocity of entrance

θ = angle between tangent to wheel and the vane at entrance

If the air is to enter without shock, the circumferential velocity u_1 must be equal to $u_1 = v_1 \cot \theta$. This condition makes the velcities u_1 and v_1 perpendicular. Completing the velocity polygon at entrance, we obtain the relation

$$c_1^2 = v_1^2 - u_1^2 \quad \text{-------------------- } 1$$

The rotating vane now continually exerts an accelerating action on the air until it leaves the wheel at the outer circumference. The wheel has now moved to B^1 the absolute path described by the air is approximately as shown, being tangential to the radial velocity U_1 at entrance. At B^1 the air is expelled from the wheel with an absolute velocity v_2 which is tangential to its absolute path AB^1. This velocity is the resultant of two other velocities, namely; the outer peripheral velocity u_2 and the relative velocity c_2 with which the air moves along the last elements of the vane. If ϕ represents the angle with which the air leaves the circumference of the wheel, we have

$$c_2^2 = v_2^2 + u_2^2 - 2 v_2 u_2 \cos \phi \quad \text{------------- } 2$$

The entrance velocity v_1 is generated by the pressure of the atmosphere.

Let γ_0 = specific weight of water

γ_1 = specific weight of entering air

ξ_1 = coefficient of resistance to entrance

To generate the velocity v_1 there will be required a head of water equal to

$$(1+\xi_1)\frac{v_1^2\gamma_1}{2g\gamma_0} = (1+\xi_1)\frac{v_1^2}{2g\varepsilon_1}$$

where ε_1 is the ratio $\frac{\gamma_0}{\gamma_1}$ of the specific weights of water and air. The pressure at A is therefore less than that in the supply pipe by an amount equal to the head consumed in generating the velocity v_1. The pressure in the supply pipe, where the air is taken directly from the atmosphere, is that due to the water barometer. For a suction fan, it is smaller by an amount equal to the resistances in the suction pipe. Let h_1 be the height of a water manometer at entrance to a fan. For a blower h_1 would equal zero, and for an exhauster it would be negative.

Let x = head measuring pressure at A

b = height of water barometer

$$x = b + h_1 - (1+\xi_1)\frac{v_1^2}{2g\varepsilon_1}$$

Substitute z_1 for $\xi_1\frac{v_1^2}{2g\varepsilon_1}$ and we have

$$x = b + h_1 - z_1 - \frac{v_1^2}{2g\varepsilon_1} \qquad 3$$

Let h_2 be the height of the water manometer placed at the end of the outlet, or in the case of a suction fan where the air flows into the atmosphere. Let w be the velocity with which the air leaves the fan and let z_2 be the head measuring the resistance in the outlet. The air which leaves the wheel at B with the absolute velocity v_2 and pressure y must not only overcome the pressure $b+h_2$ at the outlet and the resistance z_2, but also to impart a velocity w to the air.

Let γ_2 = specific weight of air leaving wheel

$$\frac{\gamma_0}{\gamma_2} = \varepsilon_2$$

Then

$$y + \frac{v_2^2}{2g\varepsilon_2} = b + h_2 + z_2 + \frac{w^2}{2g\varepsilon_2} \quad \text{------------} 4$$

In fan blowers h_2 is positive, in fans that discharge directly into the atmosphere h_2 is zero. With a small difference in pressure at inlet and outlet, we may place the specific weights y_1 and y_2 equal to each other, and in place of $\varepsilon_1 = \varepsilon_2$ put simply ε

As the air passes through the wheel from A to B, its pressure is increased by centrifugal force by an amount proportional to the difference of velocity heads at A and B. This may be shown in the following way.

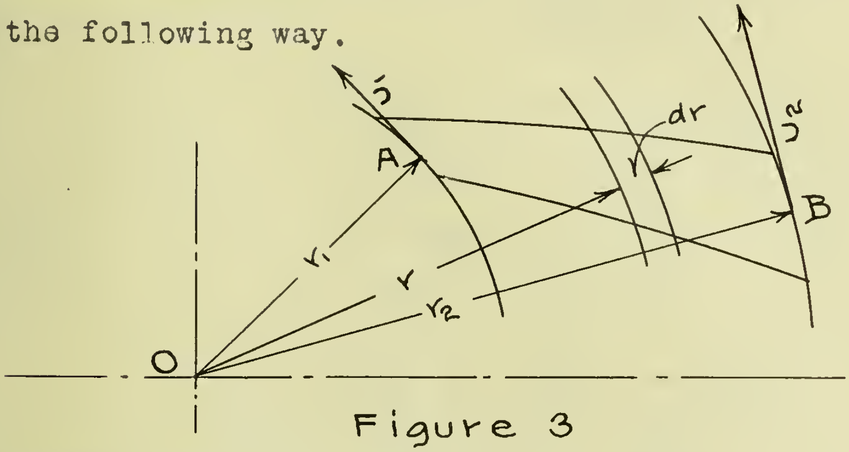


Figure 3

Consider a lamina of air as shown in figure 3 at r of cross sectional area a.

Let ω = angular velocity of channel

γ = specific weight of air

Weight of lamina $= \gamma a dr$ pounds.

Its centrifugal pressure is $\frac{\gamma}{g} a dr r \omega^2$ pounds, or changed to feet head $\frac{a\omega^2}{g} r dr$. This is the difference of pressure due to rotation.

Let dh'' = increment of pressure per unit area between r and $r + dr$

$$a\, dh'' = \frac{a\omega^2}{g} r dr$$

$$dh'' = \frac{\omega^2}{g} r dr$$

Integrating between limits r_2 and r_1

$$h_a - h_e = \frac{\omega}{g}\int_{r_1}^{r_2} r\,dr = \frac{\omega^2(r_2^2 - r_1^2)}{2g} = \frac{u_2^2 - u_1^2}{2g}$$

Here h_a = pressure head at exit.

h_e = pressure head at entrance.

This equation therefore gives us the change of pressure due to rotation. As it stands the pressure is given in feet of air, to transform this to feet of water, it is only necessary to multiply by the ratio of the densities of water and air as already shown.

As before let $\varepsilon = \frac{\gamma_0}{\gamma}$ then change of pressure measured with the water barometer is

$$\frac{u_2^2 - u_1^2}{2g\varepsilon}$$

Our last equation is now obtained from the consideration that the energy possessed by the air leaving the wheel minus the energy contained at entrance must be equal to the energy imparted by the wheel less the losses. Therefore

$$y + \frac{c_2^2}{2g\varepsilon} - \left(x + \frac{c_1^2}{2g\varepsilon}\right) = \frac{u_2^2 - u_1^2}{2g\varepsilon} - z_r \quad \text{--------} 5$$

Here z_r is the head lost in frictional resistances encountered in the wheel.

Substituting in (5) values for x and y from (3) and (4), we have:

$$b + h_2 + z_2 + \frac{\omega^2}{2g\varepsilon} - \frac{v_2^2}{2g\varepsilon} + \frac{c_2^2}{2g\varepsilon} - b - h_1 + z_1 + \frac{v_1^2}{2g\varepsilon} - \frac{c_1^2}{2g\varepsilon} = \frac{u_2^2 - u_1^2}{2g\varepsilon} - z_r$$

Recalling that $c_1^2 = u_1^2 + v_1^2$

$$c_2^2 = u_2^2 + v_2^2 - 2 v_2 u_2 \cos\beta$$

We have

$$h_2 - h_1 + z_1 + z_2 + z_r + \frac{\omega^2}{2g\varepsilon} = \frac{v_2 u_2 \cos\beta}{g\varepsilon}$$

Let $h_2 - h_1 = h$

$$z_1 + z_2 + z_r = z$$

we obtain

$$g\varepsilon(h+z)+\frac{w^2}{2} = v_2\, u_2 \cos\beta \quad\text{———————— 6}$$

From the velocity polygon at exit, by the law of sines

$$v_2 = u_2 \frac{\sin\phi}{\phi} \quad\text{———————— 7}$$

Eliminating v_2 from (6) by means of (7)

$$u_2 = \sqrt{\left(g\varepsilon(h+z)+\frac{w^2}{2}\right)\frac{\sin(\phi+\beta)}{\sin\phi\cos\beta}}$$

The trigonometric term under the radical may be reduced as follows:

$$\frac{\sin\phi\ \cos\beta + \cos\phi\ \sin\beta}{\sin\phi\ \cos\beta} = 1+\tan\beta\cot\phi$$

Or

$$u_2 = \sqrt{\left(g\varepsilon(h+z)+\frac{w^2}{2}\right)(1+\tan\beta\cot\phi)} \quad\text{———————— 8}$$

Equation (8) is the fundamental equation of the centrifugal fan and gives the peripheral velocity in terms of the change of pressure, the vane angle ϕ, and the exit angle β. When the vanes are radial, this equation leaves

$$u_2 = \sqrt{g\varepsilon(h+z)+\frac{w^2}{2}}$$

for in this case $\phi = 90°$

Having found the value of u_2 the velocity of efflux v_2 can be calculated from (6) and then the radial velocity found from the relation

$$v_r = v_2 \sin\beta$$

For a given capacity Q we can now find the necessary depth of wheel at the circumference. If l_1 and l_2 represent the depths at the inner and outer circumferences, we have

$$l_2 = \frac{Q}{v_r \times 2\pi r_2} = \frac{Q}{2\pi r_2 v_2 \sin\beta} \quad\text{———————— 9}$$

As the air enters the wheel radially the entrance width is given by

$$l_1 = \frac{Q}{v_1\, 2\pi r_1} \quad\text{———————— 10}$$

Or if l_1 is assumed v_1 can be found by eliminating Q from equation (9) and (10)

$$v_1 = \frac{r_2 l_2}{r_1 l_1} v_2 \sin\beta \text{ ------------------ } 11$$

The correct entrance angle is obtained by $\tan\theta = \frac{v_1}{u_1}$

Both terms being known from previous computations.

In the preceding, it has been assumed that the angles of discharge were given. If however β were not known, and the entrance velocity v_1 given, we could find another equation containing only the known terms.

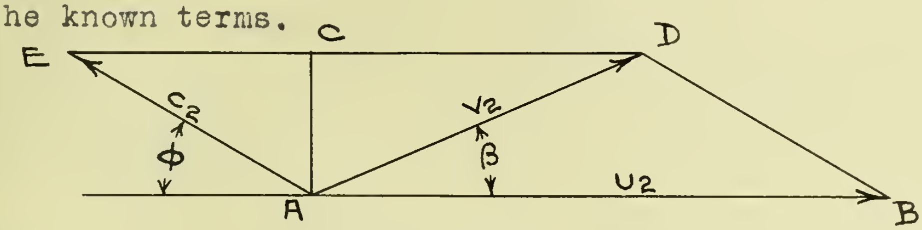


Figure 4

From figure 4

$$CD = AB - EC = u_2 - v_2 \sin\beta \cot\phi$$

combining with equation (11) and eliminating v_2

$$CD = u_2 + \frac{r_1 l_1}{r_2 l_2} v_1 \cot\phi$$

$$CD = v_2 \cos\beta$$

$$u_2 \cos\beta = u_2 + \frac{r_1 l_1}{r_2 l_2} v_1 \cot\phi$$

Substituting this value in equation (6), we have

$$\left(g\varepsilon(h+z) + \frac{\omega^2}{2}\right) = u_2\left(u_2 + \frac{r_1 l_1}{r_2 l_2} v_1 \cot\phi\right.$$

Solving this quadratic for u_2 we have

$$u_2 = \frac{r_1 l_1}{2 r_2 c_2} v_2 \cot\phi + \sqrt{\left(\frac{r_1 l_1}{r_2 l_2} v_1 \cot\phi\right)^2 + \left(g\varepsilon(h+z) + \frac{\omega^2}{2}\right)} \text{ ---- } 12$$

For radial vanes i.e. $\phi = 90°$ this equation reduces to the same form given before. Equation (12) shows that for other things being equal the peripheral velocity u_2 becomes less the larger the angle ϕ is chosen. For a given difference of pressure, the greatest circumferential velocity is obtained with vanes curved back $\phi < 90°$, and the smallest velocity with vanes radial or curved forward $\phi > 90°$. Radial Vane fans have been in common

use for many years. Recently, however, fans with blades bent forward have been placed on the market and are now in extensive service. In fact, the "Sirocco patent" is based upon this principle. Curving the blades forward permits the production of fan pressures at moderate rotative speeds.

If Q denoted the capacity in cubic feet per second delivered by the fan, and y the weight of a cubic foot, then the work expended per second is

$$W = Qy\left[h + z + \frac{\omega^2}{2g}\right] \qquad (13)$$

In this equation h and z are in feet of air. If, however, h and z are measured by the water barometer

$$W = Q\varepsilon y\left(h + z + \frac{\omega^2}{2g\varepsilon}\right) \qquad (14)$$

The useful work per second is $W_o = Qy\varepsilon h$ ------------ 15

The efficiency is therefore given by the equation

$$\eta = \frac{W_o}{W} = \frac{Qy\varepsilon h}{Qy\varepsilon\left[h + z + \frac{\omega^2}{2g\varepsilon}\right]} = \frac{h}{h + z + \frac{\omega^2}{2g\varepsilon}} \qquad (16)$$

Equation (16) calls for careful consideration. Here the efficiency has been obtained by dividing the useful work by the work expended. The result is a quotient of the head produced by the fan divided by the same quantity plus the head due to velocity of discharge and all of the losses in the fan. This efficiency is known as "The efficiency based upon static pressure." From the equation, it will be observed that η can be increased by diminishing the velocity of efflux. In the case of a fan exhausting air from a mine, and discharging it into the atmosphere, this is accomplished by the use of an expanding chimney, so that the velocity of discharge is gradually reduced. Where a fan is used for ventilation or for forced draft, the discharge velocity is limited by the size of the duct. In order to pass the large quantities of air often

required, the duct velocities may become as large as 70 ft./sec. This would correspond to pressure of 1.11 inches of water. The effect of a change in the velocity upon the efficiency depends largely upon the value of h . In pressure blowers such as are used for cupola or gas works service, where the static pressures vary from twelve to forty-five inches of water, a slight change in the velocity will have a small influence upon the efficiency. If the static pressure were twenty inches of water, and the pipe velocity 66 ft./sec. a reduction of w to almost zero would increase the efficiency less than 3%. For example take the following data:

$$h = 20$$

$$z = 10$$

$$\frac{w^2}{2g} = 1$$

$$n = \frac{20}{31} = 64\%$$

Let $w = 0$ Then

$$n = \frac{20}{30} = 66\ 2/2\%$$

In the case of volume blowers when the static pressure varies from one-half to five inches of water, a slight change in w will have a considerable influence upon the efficiency. To illustrate with an actual example, we will take a blower supplying air for heating a building. The average resistance for this service may be taken as one and one-half inches of water. The duct velocity may be assumed equal to that produced by a pressure of 1.2 inches of water.

$$h = 1.5$$

$$z = .8$$

$$\frac{w^2}{2g} = 1.2$$

In the actual case, the efficiency is

$$n = \frac{1.5}{1.5 + .8 + 1.2} = 43\%$$

If $w = 0$ then

$$h = \frac{1.5}{1.5+.8} = 65\%$$

This example serves to show to what a large extent the value of the velocity of efflux can influence the efficiency. It also emphasizes the importance of employing expanding chimneys wherever service conditions permit. It is impossible of course to make w equal to zero, but even doubling the area of the outlet would have a marked influence. Expanding chimneys are sometimes designed with an exit velocity as low as $\frac{1}{8}\sqrt{gH}$, where H is in feet of air. Substituting the values used above in this expression, we obtain 7.3 ft./sec., as the velocity of discharge. This, however, may be considered as the lower limit of the value of w .

With this method of obtaining the efficiency, if all losses were reduced to zero, the mechanical effect would still be small. Taking the data of the last illustration

Let $z = 0$ (no losses)

Then $h = \frac{1.5}{1.5+1.2} = 55.5\%$

This low value results after assuming an ideally perfect machine, the efficiency of which should be unity.

The foregoing results lead us to doubt whether the equation derived for η is a fair one to use, especially for volume blowers where the change of pressure is small. Let us compare this method with the one in general use for finding the efficiency of a centrifugal pump. To make the analogy more complete, let us assume that the water enters the pump on a level with the shaft and without pressure. With this condition of operation no useful work has been done by bringing the water into the impeller because its elevation has not been changed. A suction

gage placed at the entrance would not register zero, but would indicate a suction head equal to the velocity head of the water in the suction pipe, plus all of the losses incurred from the source to the point where the reading is taken. In finding the total head against which the pump is operating, the reading of the suction gage is added to the discharge. In this way the pump is credited with the work necessary to bring the water into the impeller. The suction pressure in a blower trial is rarely ever measured. The inlet velocity is usually above 30 ft./sec. and may be as high 60 ft./sed. If the suction head necessary to produce this velocity were added to the pressure head produced by the blower, as is done with the centrifugal pump, a much higher value for the mechanical efficiency would be obtained, as it is impracticable in most blowers to measure the depression at entrance and because the inlet and the discharge velocities are often nearly equal, the head due to the discharge velocity is sometimes added to the static pressure for determining the efficiency. The equation for n now takes the form

$$\eta = \frac{h + \frac{w^2}{2g}}{h + z + \frac{w^2}{2g}} \qquad 17$$

This value is known as efficiency based upon total dynamic pressure. In pressure blower work equation (16) is usually employed for finding the efficiency, but for volume blowers equation (17) is often given, and will be used throughout this discussion.

Referring again to the example previously given for a volume blower used for heating where

$h = 1.5$

$z = .8$

$\frac{w^2}{2ge} = 1.2$

By equation (16) the efficiency was 43%; by equation (17)

$$\eta = \frac{1.5 + 1.2}{1.5 + .8 + 1.2} = 77.1\%$$

This large difference in the two values clearly shows that the efficiency obtained from a blower test is of little value unless the method by means of which it has been obtained is fully outlined.

An entirely original and a very noteworthy contribution to the theory and design of blowers has been made by Charles H. Innes, Lecturer on Engineering at Rutherford College. This theory together with the results of a large number of blower tests with elaborate discussions and comparisons between theory and experiment has been published in his book "The Fan."

That part of the theory which is directly applicable to the problem already proposed will be included here. In this method of attack, all of the losses incurred are mathematically calculated by the aid of suitable coefficients, the value of which are obtained from experiments.

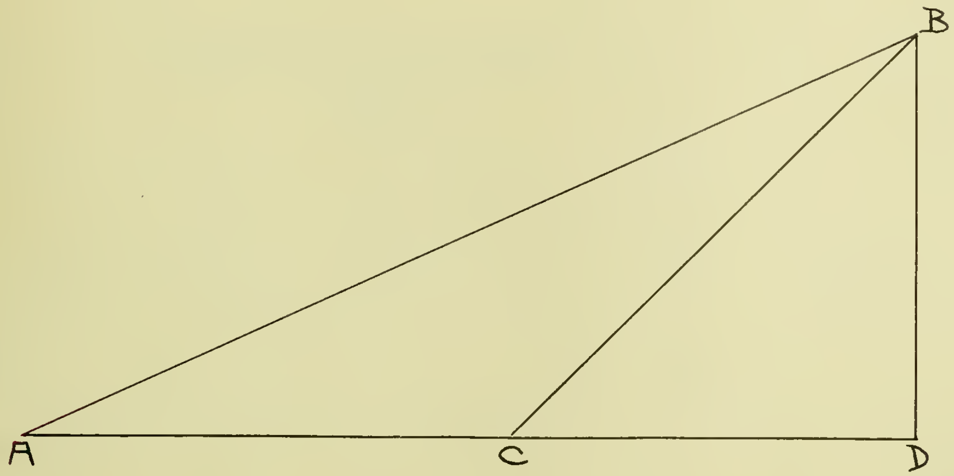


Figure 5.

When a sudden change of direction of velocity takes place, as from AB to AC, as shown in figure 5 , the loss of head will be $\frac{\overline{BC}^2}{2g}$

Let $AB =$ velocity v

$AC =$ velocity v_1

Angle $BAC = \theta$

$h =$ head

Then $h = \frac{\overline{BC}^2}{2g}$

$$= \frac{v^2 + v_1^2 - 2vv_1\cos\theta}{2g} \qquad \text{———— ———— ———— ——18}$$

By inspection of figure, it is evident that the value of h will be a minimum, when BC coincides with BD, a perpendicular to

AC. In this case $v_1 = V\cos\theta$ — — — — — — — 19

Let p = pressure before change, and p_1 after:

Then $\frac{p_1}{\gamma} + \frac{v_1^2}{2g} + \text{loss of head} = \frac{p}{\gamma} + \frac{V^2}{2g}$

$\frac{p_1 - p}{\gamma}$ = gain in pressure head

Gain in pressure head $= \frac{V^2 - v_1^2}{2g} - \frac{V^2 + v_1^2 - 2Vv_1\cos\theta}{2g}$

$$= \frac{Vv_1\cos\theta - v_1^2}{g}$$

$$= \frac{v_1(V\cos\theta - v_1)}{g}$$

$$= \frac{AC \times CD}{g} \quad \text{— — — — — 20}$$

This expression is a maximum when AC = CD, so that when a sudden change of direction must take place, and it is intended to convert a large part of the velocity head into pressure head by gradually expanding the pipe, v_1 should equal $V\cos\theta$, but where no expanding pipe can be fitted AC must be $\frac{1}{2}$AD, and hence

$$v_1 = \frac{1}{2}V\cos\theta \quad \text{— — — — — — — — 21}$$

When a sudden reduction of velocity takes place from v to v_1 without change of direction, the loss of head is

$$h = \frac{(v - v_1)^2}{2g} \quad \text{— — — — — — — — — 22}$$

The gain in pressure is found as follows:

$$\frac{p_1 - p}{\gamma} = \frac{V^2 - v_1^2}{2g} - \text{loss of head}$$

$$= \frac{V^2 - v_1^2}{2g} - \frac{(v - v_1)^2}{2g} = \frac{Vv_1 - v_1^2}{2g} \quad \text{— — — — 23}$$

Before proceeding with the derivation of the equation of a fan, we must first introduce the law of change of moment of momentum which has an important bearing upon the theory governing the action of fans.

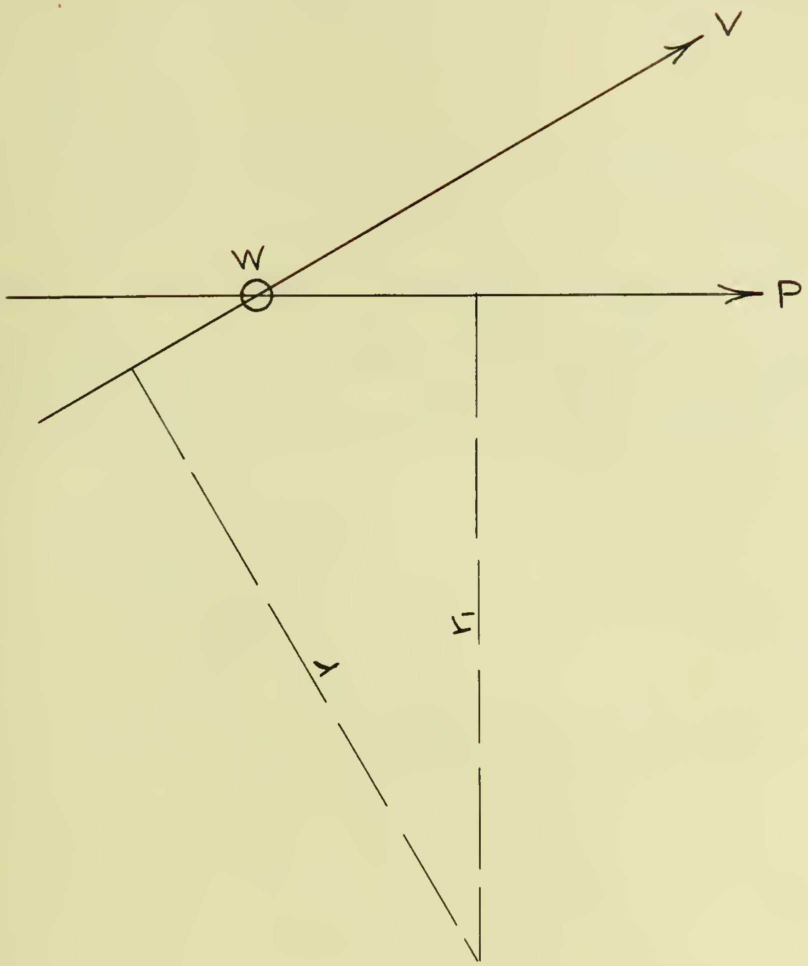


Figure 6

Let W = weight of body

v = its velocity

p = given force

Then $\frac{Wv}{g}$ = momentum of body, and

$\frac{Wvr}{g}$ = moment of momentum.

Let the force P act for a small interval of time t, and produce a velocity V in the direction of P. As impulse is equal to increase in momentum

$$\frac{WV}{g} = Pt$$

If r_1 is the perpendicular on P's direction, then $P\,t\,r_1$ is the moment of the impulse of the force, or its angular impulse; and the change of the moment of momentum is $\frac{WV}{g} r_1$. It now follows: That the change of the moment of momentum of a mass acted upon by forces is equal to the moment of the impulse of the external forces.

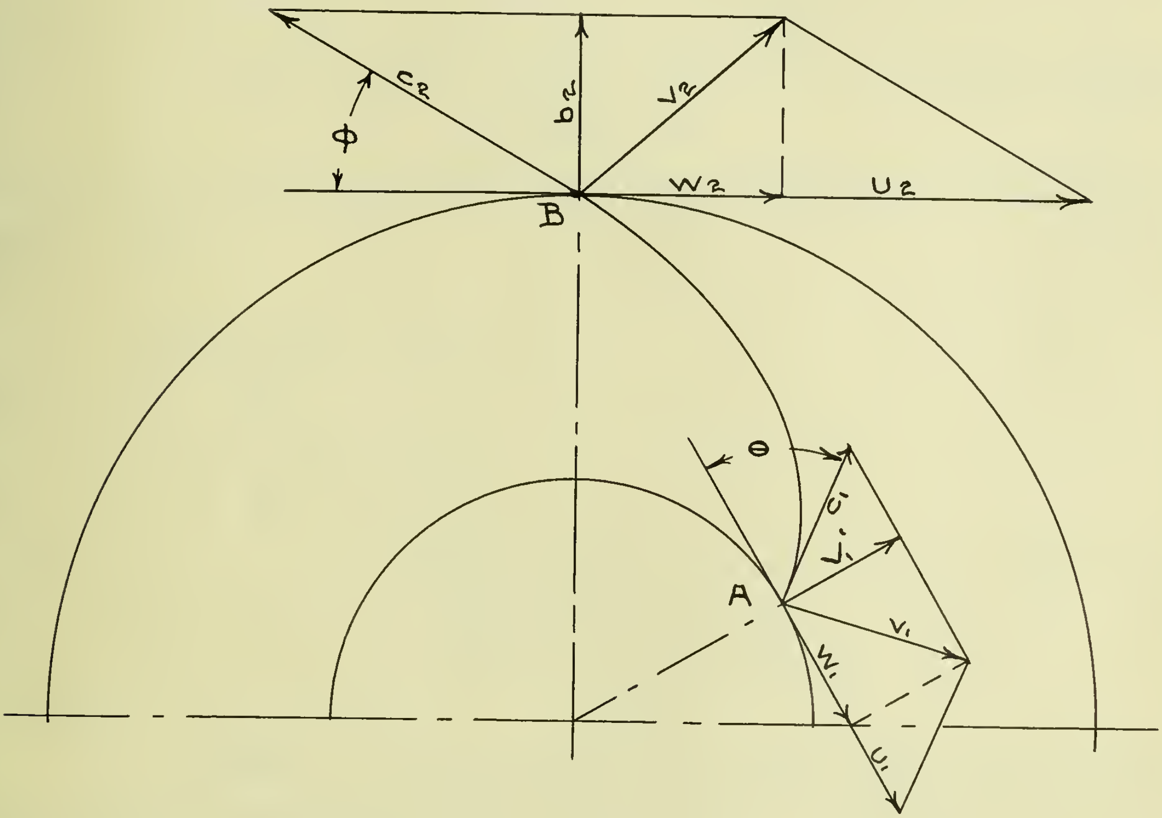


Figure 7.

Imagine a mass of air passing through a fan wheel as shown in figure 7, rotating about the axis O. Let all particles follow

paths similar to AB, and let inflow be radial.

Using the same notation as before, let

u_2 = velocity of outer circumference

u_1 = " " inner circumference

r_2 = outer radius

r_1 = inner radius

v_2 = absolute velocity of discharge

v_1' = inflow before vanes act upon each particle

v_1 = absolute velocity just after

ω = angular velocity

Since a particle of air in passing through the wheel had no moment of momentum before it entered, and as $\frac{W_1 w_2 r_2}{g}$ is its moment of momentum, after leaving, therefore:

$$\frac{W_1 w_2 r_2}{g} = \text{angular momentum of all forces acting on the particles.}$$

If W is the total weight of air passing through the wheel per second, then

$$\frac{W w_2 r_2 \omega}{g} = T\omega$$

$$\frac{W w_2 u_2}{g} = T\omega$$

= work per second, if T is the twisting moment in foot pounds.

Therefore:

$$\frac{w_2 u_2}{g} = \text{work done by the wheel per pound of air} \tag{24}$$

If we neglect the friction of the bearings, this expression is the work done on the fan shaft, irrespective of the way the air approaches the blower. If no force acts upon the air, before it reaches the fan, it can have no moment of momentum, and therefore must

approach the fan radially or axially. Equation (24) not only applies to radial flow fans, but also to those in which the flow is changed from an axial to a radial direction, known as mixed flow fans.

In passing through a blower, the air suffers several losses of head which, by proper design, may be entirely avoided, or reduced to a minimum when operating against a resistance equal to the equivalent orifice for which it was intended.

Referring to figure 7, we note that the direction of the entering air at A may be suddenly changed from v_1' to v_1; the loss of head is

$$L_1 = \frac{(v_1' - v_1)^2}{2g}$$

$$= \frac{(u_1 - c_1 \cot\theta)^2}{2g}$$

To avoid this loss at entrance, we must make the angle θ of such magnitude that

$$\cot\theta = \frac{u_1}{c_1}$$

When this relation obtains the expression for L_1 vanishes, this however can only be for one value of $\frac{Q}{\sqrt{gH}}$. After passing through the wheel, the air may enter either the diffuser, volute, or atmosphere. In the last case, the head lost is $\frac{v_2^2}{2g}$, as the kinetic energy at discharge is all lost. There is no loss entering the diffuser while in entering the volute the loss is from figure 8

$$L_2 = \frac{(v_2 v_4)^2}{2g}$$

$$= \frac{b_2^2 + (w_2 - v_4)^2}{2g} \qquad \text{— — — — — 25}$$

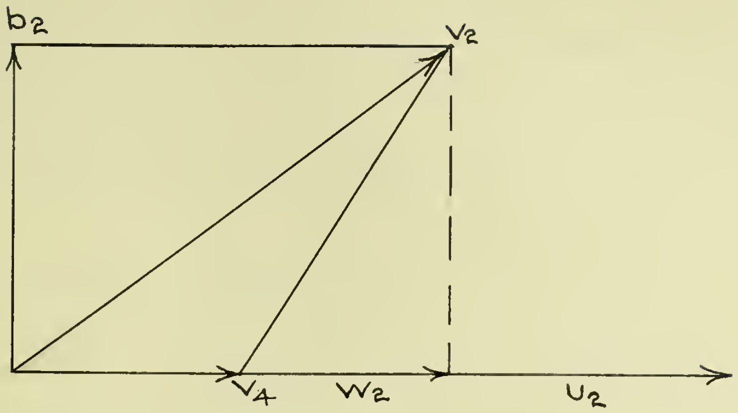


Figure 8

The velocity in the volute is v_4 which has a direction nearly tangential to the wheel. If the fan has no chimney nor expanding discharge pipe, v_4 should be equal to $\frac{1}{2}w_2$, but if it has, then v_4 should equal w_2 (See equations 20 and 21)

Assume that the fan has a diffuser, as shown in figure 9 with BD for its

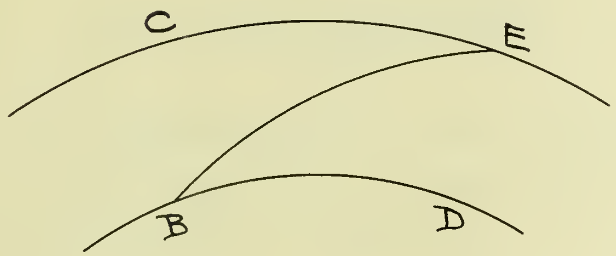


Figure 9

inner circumference, and CE its outer circumference. Let the radius of the latter be r_3. There is no change in the angular momentum of any particle, because no force acts on it during its

passage through the diffuser.

Let b_3 = radial component of discharge from diffuser

w_3 = tangential component of discharge from diffuser

a_2 = breadth at inflow

a_3 = breadth at discharge

Then

$$w_2 r_2 = w_3 r_3$$

$$\frac{w_3}{w_2} = \frac{r_2}{r_3} \quad \text{———} \quad 26$$

And

$$2\pi r_2 a_2 b_2 = 2\pi r_3 a_3 b_3$$

$$\frac{b_3}{b_2} = \frac{a_2 r_2}{a_3 r_3} \quad \text{———} \quad 27$$

If the sides are parallel, then $a_2 = a_3$

And

$$\frac{b_3}{b_2} = \frac{r_2}{r_3} \quad \text{———} \quad 28$$

In this case, the path BE is an equiangular spiral.

The air next passes into the volute, and the loss is as before,

$$L_3 = \frac{b_3 + (w_3 - v_4)^2}{2g} \quad \text{———} \quad 29$$

and if there is no expanding pipe, then v_4 should equal $\frac{1}{2} w_3$, and if there is, v_4 should equal w_3, as previously shown. The loss due to surface friction may be written

$$L_4 = F_1 \frac{c_2^2}{2g} + F_2 \frac{v_4^2}{2g} \quad \text{———} \quad 30$$

where F_1 and F_2 are suitable constants depending upon the proportions of the fan.

When operating at the proper orifice, we can substitute for the value of the velocity v_4 its equivalent in terms of w_2

and w_3. Then

$$L_2 = \frac{b_2^2}{2g} \quad \text{or} \quad \frac{b_2^2 + \frac{W_2^2}{4}}{2g} \text{— — — — —} \quad 31$$

$$L_3 = \frac{b_3^2}{2g} \quad \text{or} \quad \frac{b_3^2 + \frac{W_3^2}{4}}{2g} \text{— — — —} \quad 32$$

The work done by the wheel per second is equal to the head H multiplied by the weight of air per second plus the work absorbed by losses of head. Therefore

$$\frac{w_2 W_2}{g} \text{ — losses of head} = H$$

If the fan has no casing, the losses are

Loss at discharge

Loss due to surface friction $= L_4$

Loss at entrance $= L_1$

Then

$$\frac{W_2 U_2}{2g} - \frac{V_2^2}{2g} - F_1\frac{c_2^2}{2g} - \frac{(u_1 - c_1\cot\theta)^2}{2g} = H \text{ — — —} \quad 33$$

$$W_2 = U_2 - b_2 \cot\phi$$

$$V_2 = b_2^2 + W_2^2 = b_2^2 + (U_2 - b_2\cot\phi)^2$$

$$c_2^2 = b_2^2 \operatorname{cosec}^2\phi = b_2^2(1 + \cot\phi)^2$$

$$U_1 = U_2\frac{r_1}{r_2}$$

With these relations, it is possible to transform equation

33 into a form containing the peripheral velocity, the head and other constants made up of the frictional coefficients and the vane angles.

If the fan has a diffuser but no volute, then

$$\frac{W_2 U_2}{g} - L_1 - L_4 - \frac{V_4^2}{2g} = H$$

Substituting the expressions already obtained for these losses, we have

$$\frac{U_2 W_2}{g} - \frac{(U_1 - c_1 \cot\phi)^2}{2g} - F_1 \frac{c_2^2}{2g} - F_2 \frac{V_4^2}{2g} - \frac{V_4^2}{2g} = H$$

But $V_4^2 = W_3^2 + b_3^2$

Therefore

$$\frac{U_2 W_2}{g} - \frac{(U_1 - c_1 \cot\phi)^2}{2g} - F_1 \frac{c_2^2}{2g} - \frac{W_3^2 + b_3^2}{2g}\left[F_2 + 1\right] = H \qquad 34$$

If the fan has a volute and chimney, but no diffuser, then

$$\frac{U_2 W_2}{2g} - L_1 - L_2 - L_4 - \frac{V_5^2}{2g} = H$$

Substituting the proper expression for the losses, we have

$$\frac{U_2 W_2}{2g} - \frac{(U_1 - c_1 \cot\phi)^2}{2g} - \frac{b_2^2 + (W_2 - V_4)^2}{2g}$$

$$- F_1 \frac{c_2^2}{2g} - F_2 \frac{V_4^2}{2g} - \frac{V_5^2}{2g} = H \qquad 35$$

Lastly, if there is a volute chimney and diffuser

$$\frac{u_2 w_2}{g} - L_1 - L_2 - L_4 - \frac{v_5^2}{2g} = H$$

Substituting, as before

$$\frac{u_2 w_2}{g} - \frac{(u_1 - c_1 \cot\theta)^2}{2g} - \frac{b_3^2 + (w_3 - v_4)^2}{2g} - F_1\frac{c_2^2}{2g} - F_2\frac{v_4^2}{2g} - \frac{v_5^2}{2g} = H \qquad 36$$

In all of these characteristic equations, the velocities may be expressed in terms of u_2, the angles θ and ϕ and the discharge Q.

The mechanical efficiency of a fan does not differ largely from the ratio of the useful work divided by the work done by the wheel, expecially when operating at or near the equivalent orifice for which it was designed. Toward shut off the friction of the bearings forms a much larger percentage of the lost work and then the above definition for efficiency is not strictly true. If n represents the mechanical efficiency, then

$$n = \frac{\text{useful work per pound of air}}{\text{work of wheel per pound of air}}$$

useful work per pound $=$ H

work of wheel per pound $= \dfrac{u_2 w_2}{g}$

Therefore $n = \dfrac{gH}{u_2 w_2}$ —— —— —— —— 37

In reducing the characteristic equations for a fan, we shall consider that inflow takes place without shock, or

$$u_1 = c_1 \cot\theta$$

At this point, we shall take $b_2 = .5\sqrt{gH}$

Substituting the values found for w_2, v_2, and c_2 in equation 35, we have

$$\frac{u_2(u_2-b_2\cot\phi)}{g} - \frac{b_2+(u_2-b_2\cot\phi)^2}{2g} - F_1\,\frac{b_2^2\operatorname{cosec}^2\phi}{2g} = H$$

$$u_2^2 - u_2b_2\cot\phi - \tfrac{1}{2}b_2^2 - \tfrac{1}{2}u_2^2 + u_2b_2\cot\phi - \tfrac{1}{2}b_2^2\cot\phi$$

$$-\tfrac{1}{2}F_1b_2^2\operatorname{cosec}^2\phi = gH.$$

$$u_2^2 - b_2^2(1+\cot\phi) - b_2^2F_1\operatorname{cosec}^2\phi = 2gH$$

$$u_2^2 - b_2^2\operatorname{cosec}^2\phi\,[1+F_1] = 2gH$$

$$u_2^2 - [1+F_1]\,\tfrac{1}{4}\,gH\operatorname{cosec}^2\phi = 2gH \qquad\text{———— 38}$$

If the values of the coefficient F_1 is known, the mechanical efficiency can now be calculated for different values of the angle ϕ.

When there is a diffuser with parallel sides, but no volute, equation 34 may be reduced as follows: Let the external radius of the diffuser $r_3 = kr_2$ then by the relation previously shown

$$w_3 = \frac{w_2}{k} \quad\text{and}\quad b_3 = \frac{b_2}{k}$$

This gives (from 34)

$$\frac{u_2^2 - u_2b_2\cot\phi}{g} - F_1\,\frac{b_2^2\operatorname{cosec}^2\phi}{2g}$$

$$-\ \frac{b_2^2 + (u_2^2 - b_2^2\cot\phi)^2}{2k^2g}\,[1+F_2] = H$$

$$2u_2^2 - 2u_2 b_2 \cot\phi - F_1 b_2^2 \operatorname{cosec}^2\phi$$

$$- \frac{1}{k^2}[1+F_2]\left[b_2^2 + u_2^2 - 2u_2 b_2 \cot\phi + b_2^2 \cot\phi\right] = 2gH.$$

$$u_2^2\left[2 - \frac{1+F_2}{k^2}\right] - 2u_2 b_2 \cot\phi\left[1 - \frac{1}{k^2}(1+F_2)\right]$$

$$- b_2^2 \operatorname{cosec}^2\phi\left[F_1 + \frac{1+F_2}{k^2}\right] = 2gH$$

Let $b_2 = .5\sqrt{gH}$ Then

$$u_2^2\left[2 - \frac{1+F_2}{k^2}\right] - u_2\sqrt{gH}\cot\phi\left[1 - \frac{1}{k^2}[1+F_2]\right]$$

$$- gH\,\frac{1}{4}\left[F_1 + \frac{1+F_2}{k^2}\right]\operatorname{cosec}^2\phi = 2gH \qquad 39$$

If the fan has a volute and chimney, but no diffuser, then $v_4 = w_2$ and equation 35 can be put into the form

$$2u_2^2 - 2u_2 b_2 \cot\phi - F_1 b_2^2 \operatorname{cosec}^2\phi - b_2^2$$

$$- F_2(u_2 - b_2\cot\phi)^2 - v_5^2 = 2gH$$

Let $v_5 = \frac{1}{8}\sqrt{gH}$

$$u_2^2(2-F_2) - u_2\sqrt{gH}\cot\phi[1-F_2] - gH\left[2 + \frac{1}{4} + \frac{1}{64}\right.$$

$$\left. + \frac{F_1}{4}\operatorname{cosec}^2\phi + \frac{F_2}{4}\cot^2\phi\right] = 0$$

$$u_2^2(2-F_2) - u_2\sqrt{gH}\cot\phi(1-F_2)$$

$$- gH\left[2\tfrac{17}{64} + \frac{F_1}{4}\operatorname{cosec}^2\phi + \frac{F_2}{4}\cot^2\phi\right) = 0 \qquad 40$$

If there is no chimney $v_4 = \frac{1}{2} w_2$ for maximum efficiency and equation (35) becomes

$$2u_2^2 - 2u_2 b_2 \cot\phi - F_1 b_2^2 \operatorname{cosec}^2\phi - b_2^2 - \frac{1}{4}\left[u_2 - b_2\cot\phi\right]^2$$

$$-\frac{1}{4}F_2\left[u_2 - b_2\cot\phi\right]^2 - \frac{1}{4}\left[u_2 - b_2\cot\phi\right]^2 = 2gH.$$

In this case v_5 is the velocity of discharge from the volute, and is equal to $\frac{1}{2}\left[u_2 - b_2\cot\phi\right]$

Collecting like terms and putting $b_2 = .5\sqrt{gH}$, we finally obtain

$$u_2^2\left[\frac{3}{2} - \frac{F_2}{4}\right] - \frac{1}{2}\left[1 - \frac{F_2}{2}\right]u_2\sqrt{gH}\cot\phi$$

$$-\frac{1}{4}gH\left[9 + F_1\operatorname{cosec}^2\phi + \frac{1}{2}\cot^2\phi + \frac{F_2}{4}\cot^2\phi\right] = 0 \quad \text{---} \quad 41$$

Lastly, if there is a volute, diffuser and chimney

Let $r_3 = k r_1$ Then

$$2u_2^2 - 2u_2 b_2\cot\phi - F_1 b_2^2\operatorname{cosec}^2\phi - \frac{b_2^2}{k^2}$$

$$-F_2\left[\frac{u_2 - b_2\cot\phi}{k^2}\right] - \frac{gH}{64} = 2gH.$$

$$v_4 = w_3 = \frac{w_2}{k}$$

$$u_2^2\left[2 - \frac{F_2}{k^2}\right] - 2u_2 b_2\cot\phi\left[1 - \frac{F_2}{k^2}\right]$$

$$-b_2^2\left[\frac{1}{k^2} + F_1\operatorname{cosec}^2\phi + F_2\frac{\cot^2\phi}{k^2}\right] - \frac{gH}{64} - 2gH = 0$$

Or finally

$$u_2^2\left[2-\frac{F_2}{k^2}\right] - b_2\sqrt{gH}\cot\phi\left[1-\frac{F_2}{k^2}\right]$$

$$-gH\left[\frac{1}{4b^2}+\frac{F_1\operatorname{cosec}^2\phi}{4}+\frac{F_2\cot^2\phi}{4k^2}+2\frac{1}{64}\right]=0 \quad\text{—— 42}$$

In this thesis, we shall deal entirely with a fan having a volute, but without diffuser or chimney. Characteristic equation (41) applies to this type, and it will be analysed in greater detail.

The values of the coefficients F_1 and F_2, which agree best with experimental results, are both $\frac{1}{8}$. Substituting these values equation 41 reduces to

$$1.47u_2^2-\frac{15}{32}u_2\sqrt{gH}\cot\phi-\frac{gH}{4}\left[9\frac{1}{8}+\frac{21\cot^2\phi}{32}\right]=0 \quad\text{—— 43}$$

Recalling that the mechanical efficiency

$$\eta=\frac{gH}{u_2[u_2-b_2\cot\phi]} \quad\text{and}\quad b_2=.5\sqrt{gH}$$ we may now calculate the values of n for different values of the angle ϕ.

The solution in detail for $\phi = 15^{\circ}$ is as follows:

$$\cot\phi = \cot 15^{\circ} = 3.73$$

Equation 43 becomes

$$1.47u_2^2-.48\times3.73\,u_2\sqrt{gH}-\frac{gH}{4}\left[9.125+\frac{21\times\overline{3.73}^2}{32}\right]=0$$

$$1.47u_2^2-1.79u_2\sqrt{gH}-4.556\,gH=0$$

$$u_2^2-1.21u_2\sqrt{gH}-3.1\,gH=0$$

Factoring this equation

$$\left[u_2 - 2.45\sqrt{gH}\right]\left[u_2 + 1.26\sqrt{gH}\right] = 0$$

$$u_2 = 2.45\sqrt{gH}$$

Then

$$\eta = \frac{gH}{2.45\sqrt{gH}\left[2.45\sqrt{gH} - .5\sqrt{gH} \times 3.73\right]}$$

$$= \frac{1}{2.45\left[2.45 - 1.86\right]} = 69.5\,\%$$

In the same way the efficiency for other values of the angle ϕ were computed, and the results given in the following table

$\phi = 15^\circ$	$u_2 = 2.45\sqrt{gH}$	$n =$	.69
20°	$2.\ \sqrt{gH}$	$n =$	.75
30°	$1.67\sqrt{gH}$	$n =$	.74
45°	$1.46\sqrt{gH}$	$n =$	.70
90°	$1.24\sqrt{gH}$	$n =$	.64

These values of n are based upon static pressure. To show the effect of adding a chimney upon the efficiency, we will compute the values of n from equation (40) inserting the numerical values of the coefficients F_1 and F_2.

$$1.875u_2^2 - .875u_2\sqrt{gH}\cot\phi - gH\left[2.297 + \frac{\cot^2\phi}{16}\right] = 0 \qquad (44)$$

$\phi =$ 15	u_2	=	$2.43\sqrt{gH}$	n	=	.72
30			1.61			.83
45			1.37			.83
90			1.10			.81
120			.99			.79
135			.91			.78

V -- ANALYSIS OF BLOWER TESTS.

The question now naturally arises, "Does the theory here presented compare with experimental results obtained by testing blowers?" If it can be shown that the performance of these machines bears out the theoretical analysis, we can proceed to apply it to the problem, previously proposed, and be reasonably certain that the desired results will be obtained. In bringing out a new type of fan, the designer is placed under very severe restrictions. A given pressure must be obtained at a given delivery with fixed speed. In addition to this, the efficiency should also be a maximum at the rated condition of discharge and pressure. It is essential therefore to have a clear insight into the application of existing theory and to know where it coincides with and where it departs from experimental results. This can only be obtained by analyzing published experiments made upon fans. Although the results of many blower tests have been published in the technical journals, and proceedings of engineering societies, the task of making an investigation to determine whether the theoretical laws apply is a very difficult one. In nearly all cases, only the general dimensions of the machine are given. Nothing is said of the vane angles at entrance and exit, diameter of the impeller, and such other information, which is essential for a complete analysis. In addition to this, the experimental data is not always reliable on account of the great difficulty of measuring air accurately. In many instances, the discharge is exaggerated and the fan credited with a better performance than it actually gives.

When a comparison is made between theory and experiment, discrepancies are usually found that would indicate that the fundamental analysis was wrong. It must be borne in mind that in order to develop any theory, certain assumptions must be made. If these are not fulfilled in practice, we can not expect an agreement with experiment. For example, the air should enter the wheel without obstruction, or shock. Yet, how few blowers are upon the market in which these two conditions are fulfilled. As the air is discharged from the impeller, it should leave at the vane angle. It is not at all reasonable to suppose that the air is discharged tangentially to the vane, especially when the number of vanes is small. Even with a sufficient number of blades, the channel cross section at exit may be such as to make it impossible for all of the particles to be discharged at the same angle. What actually happens in a blower is that the air slips over the ends of the vane making the actual angle ϕ smaller than the vane angle. This necessitates a higher speed than the theoretical for a given pressure.

We shall use the following tests for a comparison between theory and experiment:

Results of tests upon two impellers made by Albert E. Guy, chief engineer of the De Laval Steam Turbine Co., and reported in the American Machinist for June 29, 1911.

"Experiments upon Centrifugal Fans" by Heenan & Gilbert, reported in the Proceedings of the Institution of Civil Engineers.

"Experiments upon Centrifugal Fans" by Bryan Donkin, reported in the Institution of Civil Engineers.

While many blower tests have been made in this country practically all of them are useless, as far as scientific analysis

of the results is concerned. The tests of Albert Guy have been selected, for the reason that a drawing showing the detail construction of the impellers accompanied the report. Furthermore, the author is one of the foremost American centrifugal pump engineers, and has the distinction of developing that efficient type of pump now manufactured by the De Laval Steam Turbine Co. The runners used for these tests were carefully and scientifically designed so that conditions for a fair comparison are extremely favorable.

In these tests, the shaft horsepower was obtained by using a calibrated turbine. The air pressure was varied by changing the nozzle areas at the end of the discharge pipe. The quantity of air was measured by means of the Pitot Tube inserted in the duct.

Wheel No. 1, shown in figure 10was designed to deliver 7000 cubic feet of air per minute at a static pressure of 22 inches of water; the speed being 3600 r.p.m.

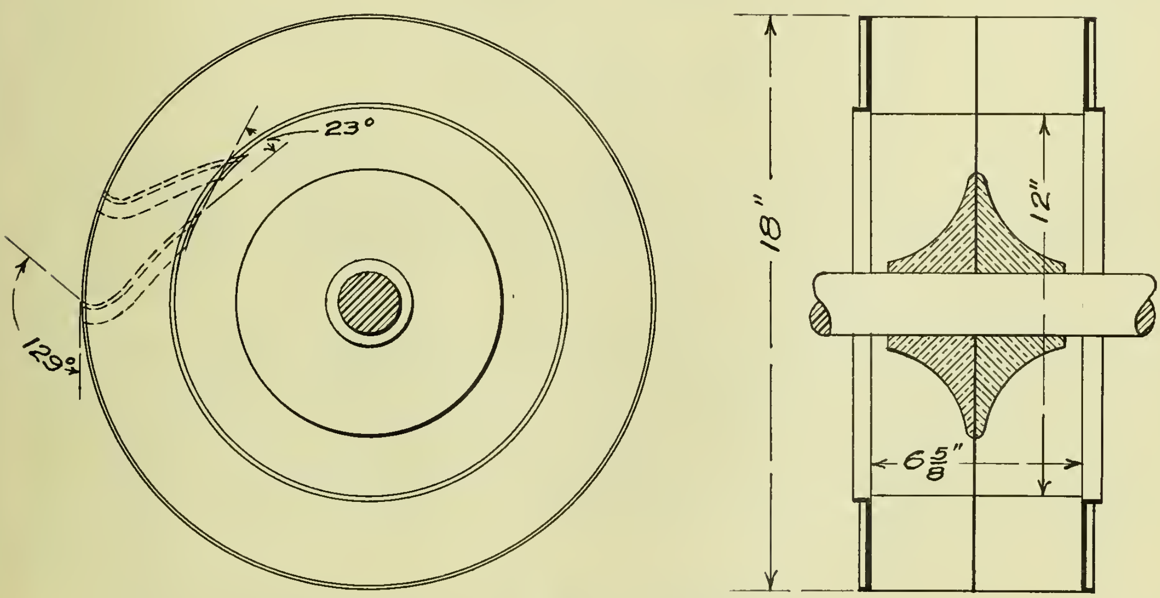


Figure 10.

Inlet Area.

Area 12 inch circle $=$ 113.1 sq. in.

" $2\frac{3}{4}$ " hub $=$ 5.94

107.16 sq. in.

Total Inlet Area $= \frac{2 \times 107.16}{144} = 1.49$ sq. ft.

Axial Inlet Velocity $\frac{7000}{60 \times 1.49} = 78$ ft./ sec.

Inlet Area to Vanes is

$$\frac{\pi \times 12 \times 6}{144} = 1.57 \text{ sq. ft.}$$

Radial inlet velocity $= \frac{7000}{60 \times 1.57} = 74$ ft./sec, neglecting the thickness of the vanes. The true value is about 78 ft./sec, the same as the axial inlet velocity.

Velocity of the vanes at entrance is

$$\frac{3600 \times \pi \times 12}{12 \times 60} = 189 \text{ ft./sec.}$$

For entrance without shock, the angle at entrance is determined by the equation

$$\cot \theta = \frac{u_1}{c_1}$$

In this case,

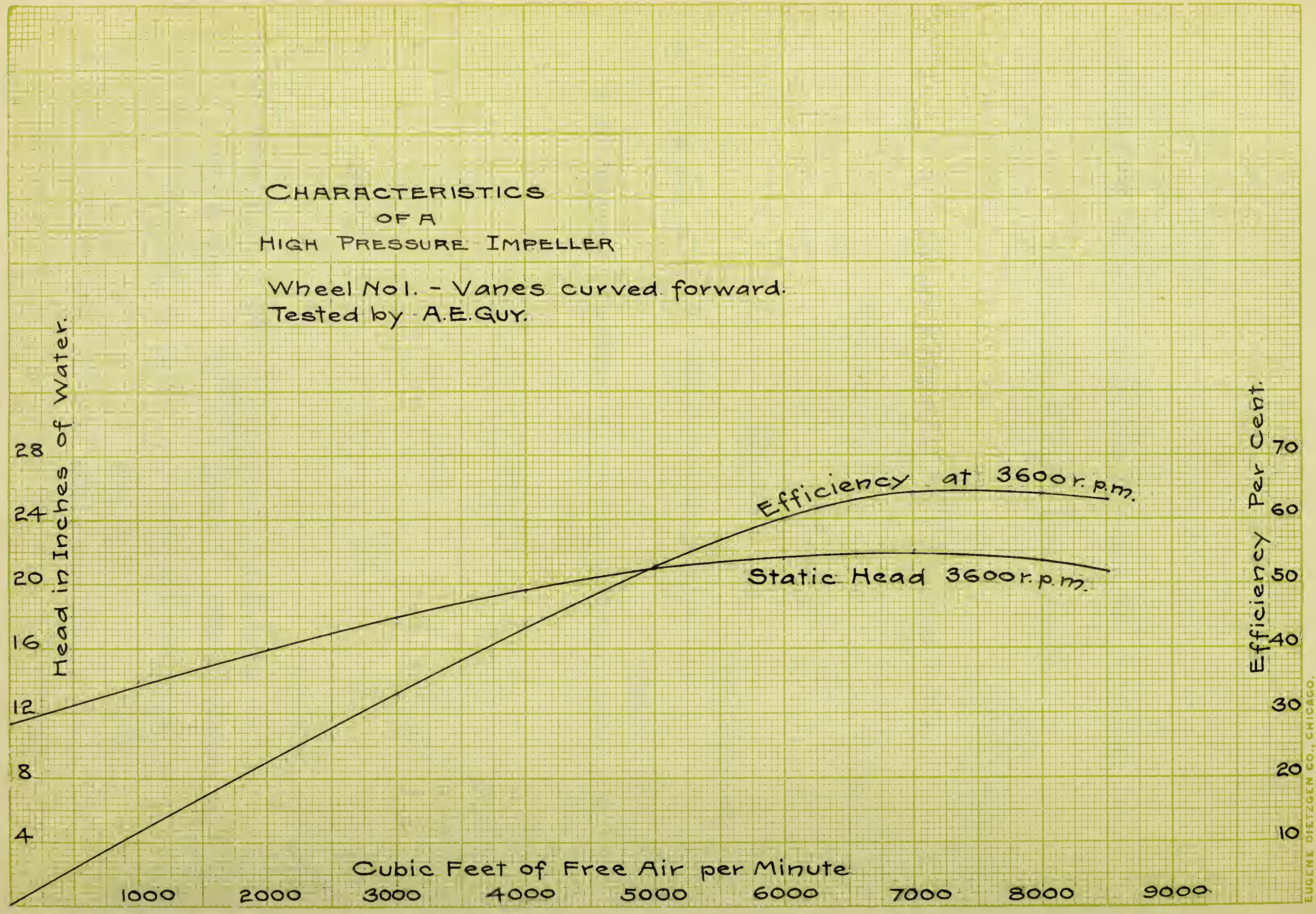
$$\cot \theta = \frac{189}{78} = 2.42$$

$$\theta = 22^\circ - 30'$$

Compared with 23°, as shown in the figure. The condition of entrance for maximum efficiency is in this case completely fulfilled.

The complete results of this test are shown graphically by the characteristic curves on page 54. At rated delivery and pressure, the mechanical efficiency based upon static pressure was 64%.

The discharge area of the impeller is



CHARACTERISTICS
OF A
HIGH PRESSURE IMPELLER
Wheel No1. - Vanes curved forward.
Tested by A.E.GUY.
Head in Inches of Water.
28
24
20
16
12
8
4
Efficiency at 3600 r.p.m.
Static Head 3600 r.p.m.
Efficiency Per Cent.
70
60
50
40
30
20
10
Cubic Feet of Free Air per Minute
1000
2000
3000
4000
5000
6000
7000
8000
9000
EUGENE DIETZGEN CO., CHICAGO.

$$\frac{\pi \times 18 \times 6}{144} = 2.36 \text{ sq. ft.}$$

The radial velocity per 1000 cubic ft. per minute is

$$\frac{1000}{60 \times 2.36} = 7.06 \text{ ft./sec.}$$

At rating radial velocity = 7 x 7.06 = 49.42 ft./sec.

Peripheral velocity $= \frac{3600 \times 18\,\pi}{12 \times 60} = 283$ ft./sec.

We can now construct the velocity polygon at discharge, as shown in figure11.

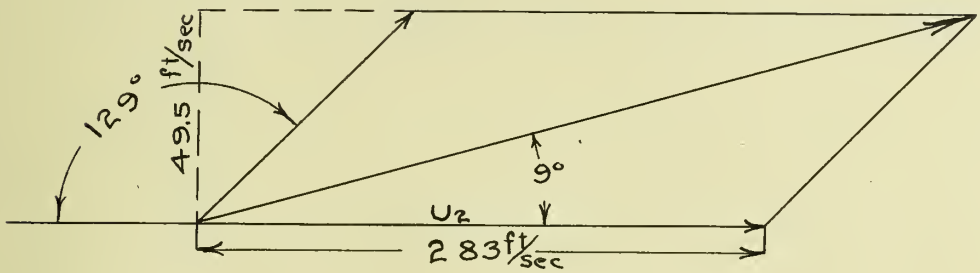


Figure 11.

The fundamental equation for the peripheral velocity of a centrifugal blower is

$$u_2 = \sqrt{\left[g\,\varepsilon(h+z) + \frac{\omega^2}{2}\right](1 + \tan\beta\ \cot\phi)} \quad \text{----8}$$

The efficiency $\quad \eta = \dfrac{h}{h+z+\frac{\omega^2}{2g\varepsilon}}$

or

$$h+z+\frac{\omega^2}{2g\varepsilon} = \frac{h}{\eta}$$

Equation (8) can now be put into the form

$$u_2 = \sqrt{g\,\varepsilon\left[h+z+\frac{\omega^2}{2g\varepsilon}\right](1+\tan\beta\ \cot\phi)}$$

Substituting

$$u_2 = \sqrt{g\,\varepsilon\,\frac{h}{\eta}}\ \sqrt{1+\tan\beta\ \cot\phi}$$

The value of e, the ratio of the pressure due to one inch of

water to the pressure due to one foot of air is determined as follows:

At 62° F one cubic foot of free air weighs .0761 lbs.

At this temperature, one cubic foot of water weighs 62.36 lbs. The pressure due to one inch of water is then $\frac{62.36}{12}$ lbs. per sq foot.

$$e = \frac{62.36}{12 \times .0761} = 68.3$$

The values of the other terms in the fundamental equation are

$$g = 32.2$$

$$h = 22$$

$$n = 64\%$$

$$\beta = 9^\circ$$

$$\phi = 129^\circ$$

$$U_2 = \sqrt{32.2 \times 68.3 \times \frac{22}{.64}}\sqrt{1 + \tan 9^\circ \cot 129^\circ}$$

$$= 276 \times .925$$

$$= 255 \text{ ft/sec.}$$

The peripheral velocity of wheel actually was 283 ft. /sec. Examining the direction of the vanes at exit, it is impossible to suppose that the entire channel of issuing air could be discharged forward at the same angle given to the vane. The curvature is close to the tip, and the air undoubtedly slips over the ends of the vane. To find the actual direction of the air, we

proceed as follows:

$$u_2 = 276\sqrt{1+\tan\beta\cot\phi}$$

$$u_2 = 283 \text{ ft/sec.}$$

Then $\sqrt{1+\tan\beta\cot\phi} = \frac{283}{276} = 1.03$

For radial vanes, $\phi = 90°$, the expression under the radical becomes equal to one. The right hand member is approximately equal to unity. This signifies therefore that the air, instead of leaving with an angle $\phi = 129°$, was discharged approximately radially with an angle $\phi = 90°$. As the radial velocity became less with decreased discharge, we would expect the air to follow more and more closely the angle of the tip of the vane. This view is borne out by the following tabular values which have been computed for the entire range of the blower's capacity.

Capacity	Static Head	ϕ	β	Radial Velocity	Eff.	Calculated Speed	Rim Speed
8500	20.2	129°	$10\frac{1}{2}°$	60	63	242	283 ft/sec
8000	21.2	"	10°	56.5	64	248	"
7000	22	"	9°	49.5	64	255	"
6000	21.3	"	$7\frac{1}{2}°$	42.5	60	263	"
5000	20.6	"	6°	35.3	53	277	"
4000	19.5	"	5°	28.2	44	296	"
3000	18	"	4°	21.2	33	332	"

The difference between the calculated and actual peripheral velocities varies as previous considerations have indicated. For the smaller capacities, however, the theoretical speed becomes

greater than the actual. In the first place, the fundamental equation is derived by considering the air to enter the wheel without shock and can only be expected to apply at the rated discharge. The efficiency as employed in this equation excludes the mechanical losses such as bearing friction. At rated load these losses form a small percentage of the total, and hence very little error is introduced by simply substituting the efficiency obtained by experiment in the fundamental equation. However, as the capacity is decreased, the purely mechanical losses form a larger and larger percentage of the total losses and consequently the efficiency substituted in the equation for peripheral velocity is too small. The result is an exaggeration of the rim speed as shown in the tabulated computation.

Wheel No. 2, shown in figure 12was designed to deliver 5250 cubic feet of free air per minute at a static pressure of five inches of water, speed 3600 R.P.M. The ganeral overall dimension of this impeller were the same as those of wheel No. 1.

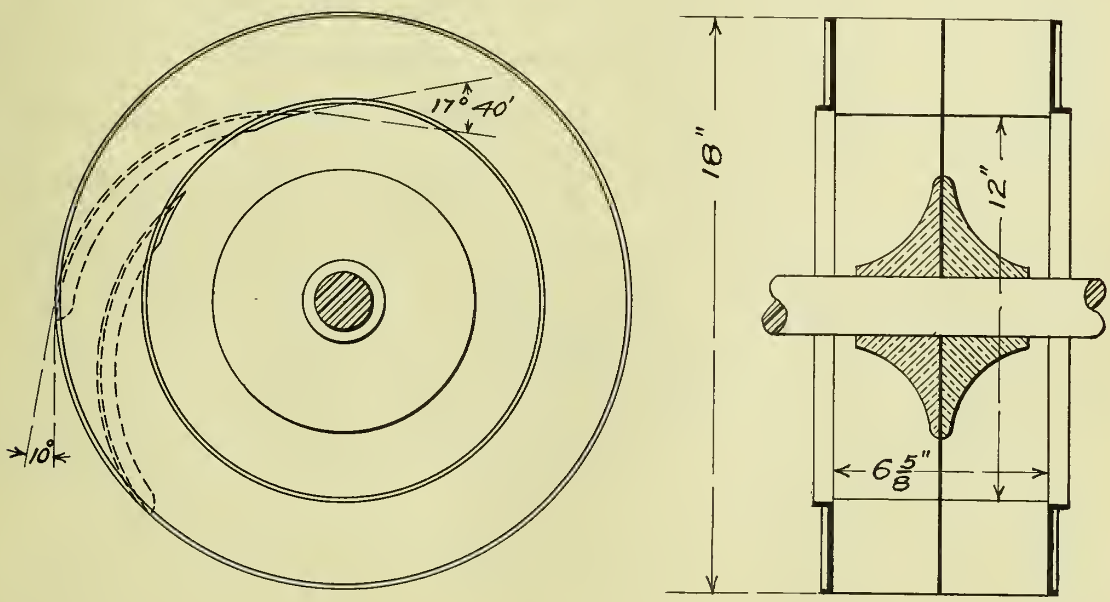


Figure 12

As before the axial inlet area $=$ 1.49 sq. ft.

Inlet velocity $= \dfrac{5250}{60 \times 1.49} = 58.5$ ft./sec

Inlet area to vanes 1.57 ft./sec.

Radial inlet velocity $= \dfrac{5250}{60 \times 1.57} = 55.8$ ft./sec, neglecting thickness of vanes.

Velocity of vanes at entrance is

$\dfrac{3600 \times \pi \times 12}{12 \times 60} = 189$ ft./sec

For entrance without shock, the angle at entrance is determined by the equation

$$\cot \theta = \frac{u_1}{c_1}$$

In this case $\cot\theta = \dfrac{189}{55.8} = 3.38$

$\theta = 16^\circ\ 30'$

The actual angle of entrance was $17^\circ\ 40'$. On account of the thickness of vanes, the radial entrance velocity would have a slightly higher value than 55.8 ft./sec, which would slightly increase the entrance angle. This clearly goes to show that this wheel was designed for radial entrance.

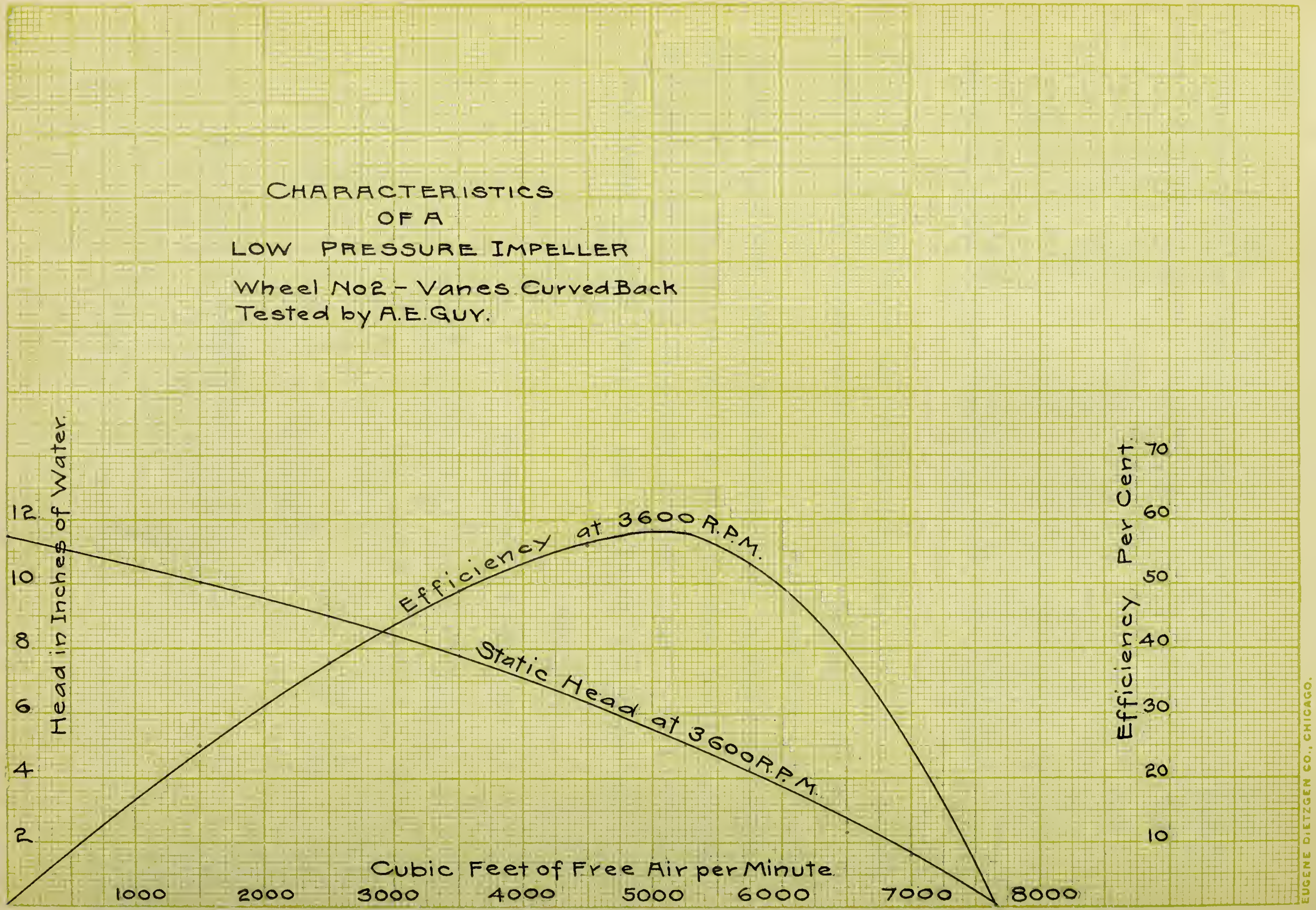
The characteristic curves on page 60 show the complete results of the test upon this wheel. At the rated delivery and pressure, the mechanical efficiency based upon static pressure was 58%.

Discharge area of impeller $=$ 2.36 sq. ft.

Radial velocity per 1000 cu. ft. per minute $=$ 7.06 ft/sec

Peripheral velocity $= \dfrac{3600 \times 18}{229} = 283$ ft./sec

With this data and the vane angle, we can construct the velocity polygon at exit as shown in figure 13


CHARACTERISTICS
OF A
LOW PRESSURE IMPELLER
Wheel No2 - Vanes Curved Back
Tested by A.E.GUY.
Head in Inches of Water.
2
4
6
8
10
12
Efficiency at 3600 R.P.M.
Static Head at 3600 R.P.M.
Cubic Feet of Free Air per Minute
1000
2000
3000
4000
5000
6000
7000
8000
Efficiency Per Cent.
10
20
30
40
50
60
70
EUGENE DIETZGEN CO., CHICAGO.

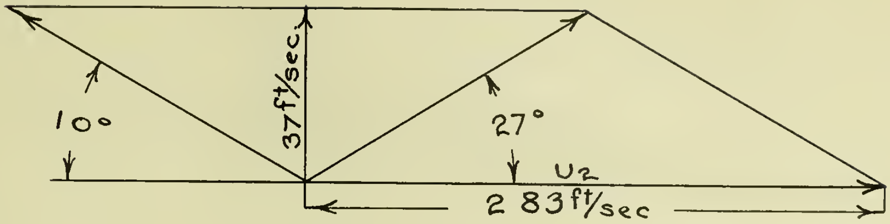


Figure 13.

$$\frac{h}{\eta} = \frac{5}{.58} = 8.62$$

$$u_2 = \sqrt{g\varepsilon\frac{h}{\eta}\left(1+\tan\beta\cot\phi\right)}$$

$$= \sqrt{32.2 \times 68.3 \times 8.62}\sqrt{(1+\tan 27^\circ \cot 10^\circ)}$$

$$= 1.97\sqrt{32.2 \times 68.3 \times 8.62}$$

$$= 270 \text{ ft/sec}$$

The actual rim speed was 283 ft./sec. In this wheel the passages have a better form for directing the air and therefore we should expect less slipping over the ends of the blades. The theoretical velocity does approach the actual much more closely than it did with wheel No. 1.

The same analysis was applied to the entire range of capacity and the results of these computations are given in the following table:

Capacity	Static Pressure	ϕ	β	Eff.	Radial Velocity	Calculated Speed	Rim Speed
5750	4.3	10°	37°	53	40.6		283
5250	5	"	27°	58	37	270	"
4500	6.3	"	17°	56	31.8	258	"
3500	7.8	"	10°	49	24.7	264	"
2500	9	"	5°	38	17.6	278	"
1500	10	"	$2\frac{1}{2}^\circ$	25	10.6	329	"

One of the most striking results of this analysis is that, although these impellers exhibit widely different characteristics and are designed for different pressures and have entirely different vanes when compared to the theoretical, show similar divergencies, none of which are very great. This in itself is one of the strongest proofs that the theory of centrifugal blowers, advanced by Weisbach, is correct.

We will next consider the experiments of Heenan and Gilbert. The general dimensions of wheel No. 1 are shown in figure 14. This impeller had six blades, bent back to form an angle of 35° with the tangent at the outer radius.

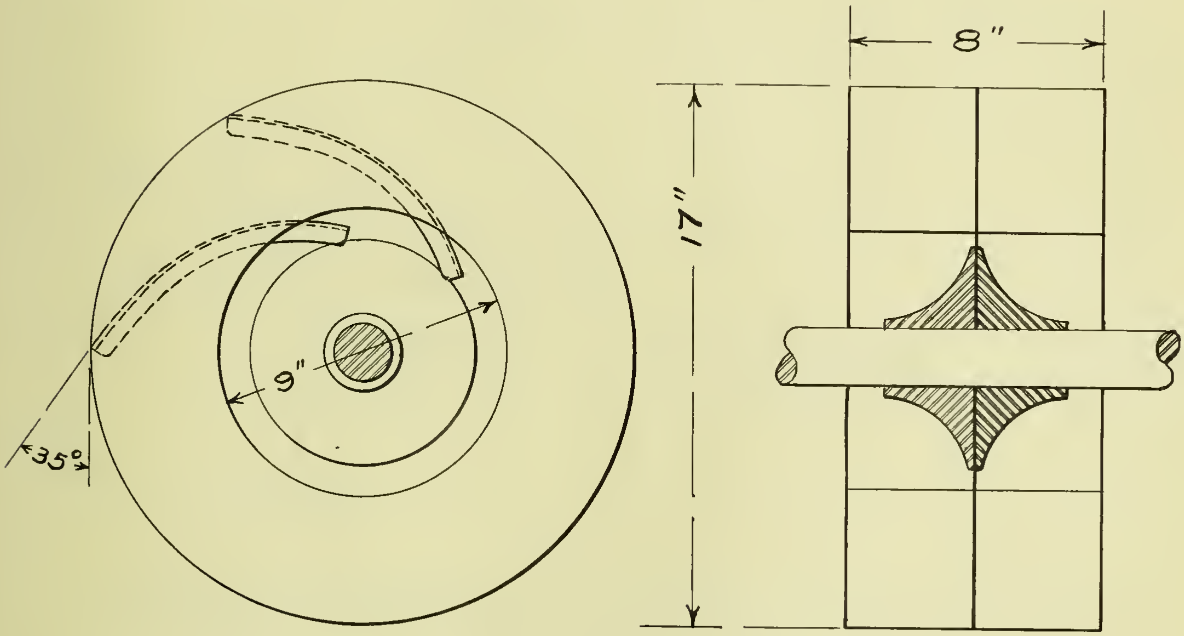


Figure 14

At a capacity of 2500 cubic feet of air per minute, with a

tip speed of 200 ft./sec. a compression of 6.8 water gage was obtained with an efficiency of 52%.

Discharge area of wheel $= \dfrac{\pi \times 17 \times 8}{144} = 2.97$ sq. ft.

Radial velocity per 1000 cubic feet per minute =

$\dfrac{1000}{60 \times 2.97} = 5.6$ ft./sec

This results in the following velocity polygon, figure 15

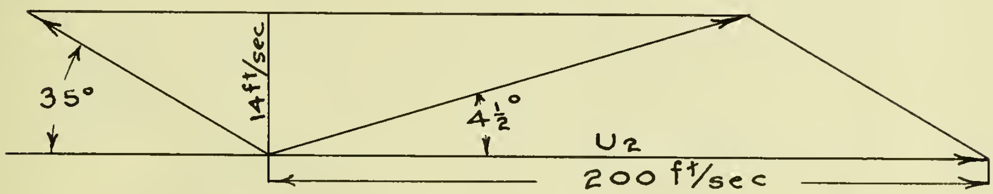


Figure 15.

$$\frac{h}{\eta} = \frac{6.8}{.52} = 13.07$$

$$U_2 = \sqrt{g\varepsilon \frac{h}{\eta}\left(1 + \tan\beta \cot\phi\right)}$$

$$= \sqrt{32.2 \times 68.3 \times 13.07}\ \sqrt{1 + \tan 4\tfrac{1}{2}^\circ \cot 35^\circ}$$

$$= 182\ \text{ft/sec.}$$

This is the theoretical rim velocity for the given compression. The actual was 200 ft./sec. For the above conditions, we should expect a rather close agreement between the actual and the theoretical peripheral velocities, because the discharge is not large and consequently the velocity polygon at exit should take a form almost equal to that indicated by the direction of

the vanes. However, the impeller only had six blades which may account in a large measure for the discrepancy obtained. Wheel No. 2 had vanes with radial tips. This form of vane is usually adopted when moderately high pressures are desired. Except for the shape of the blade, this impeller had the same general dimensions as wheel No. 1, as shown by figure 16.

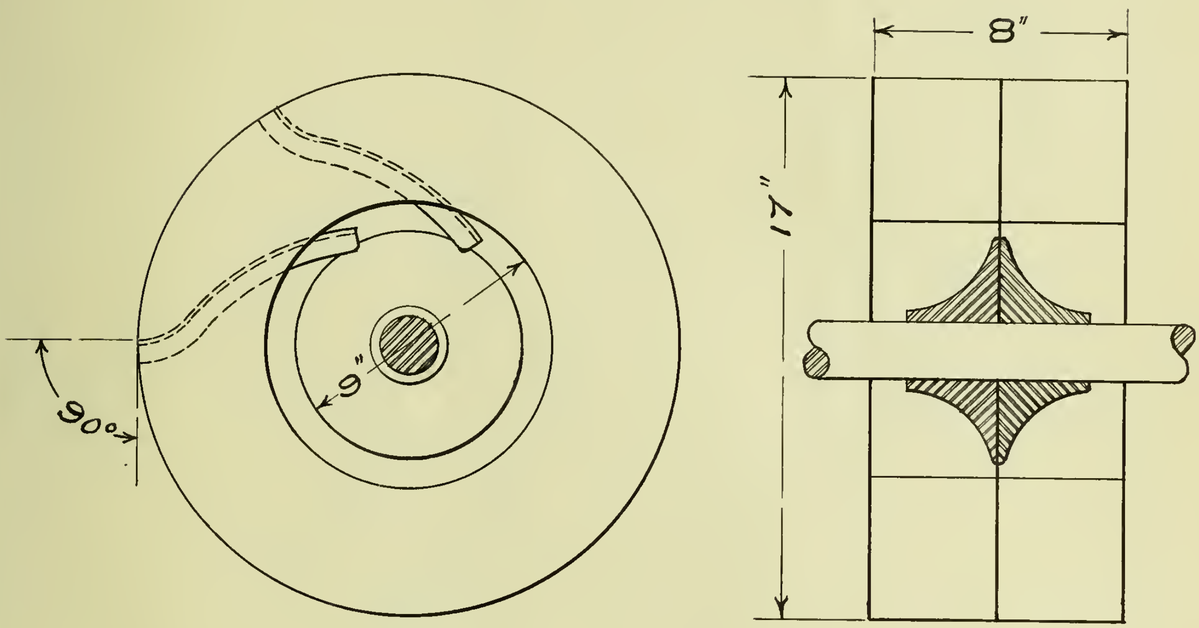


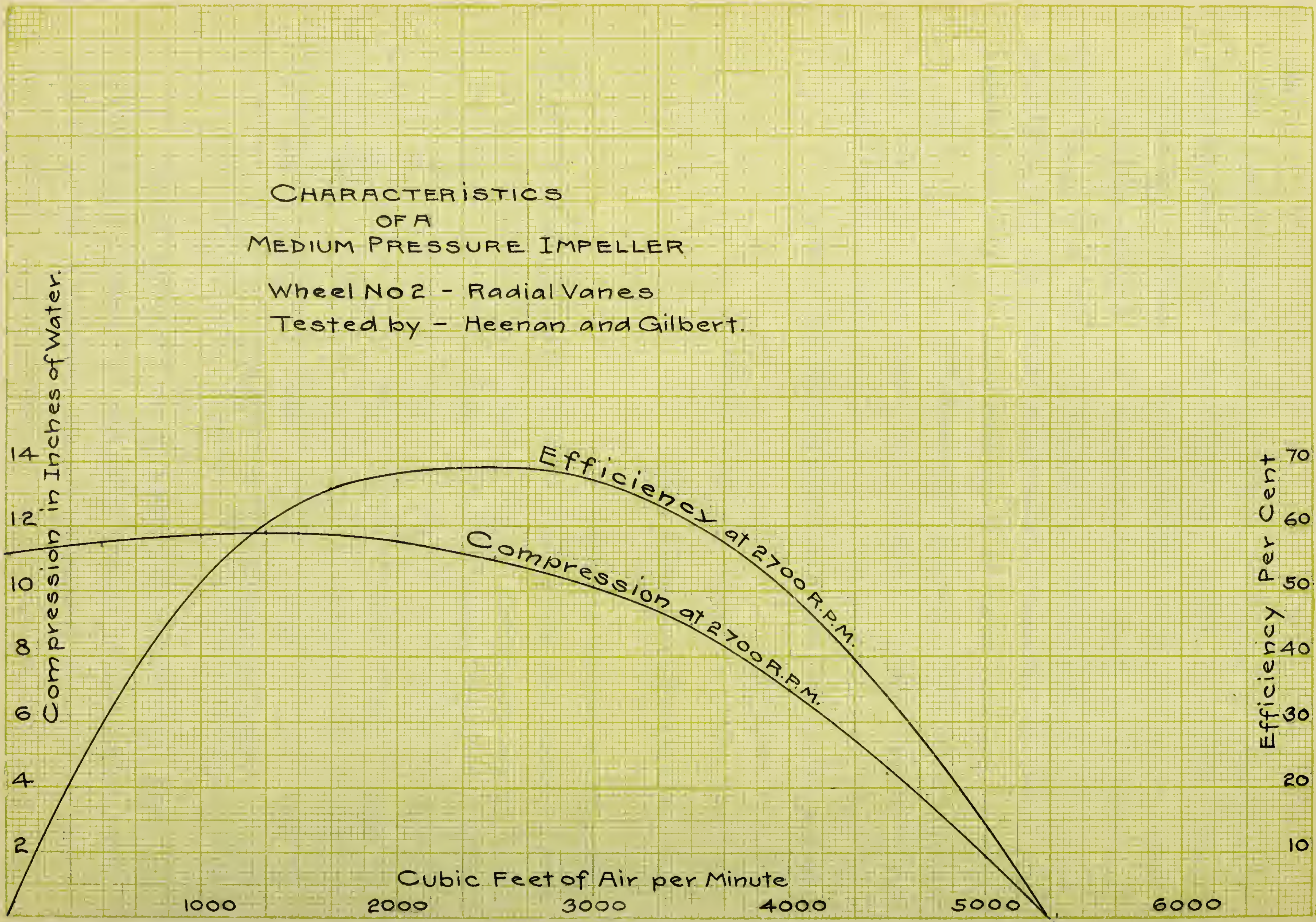
Figure 16

The results of the test upon this wheel are shown graphically on page 65.

As before, discharge area = 2.97 sq. ft.

Radial velocity per 1000 cubic feet per minute = 5.6 ft./sec

Peripheral velocity = 200 ft./sec


CHARACTERISTICS
OF A
MEDIUM PRESSURE IMPELLER
Wheel No 2 - Radial Vanes
Tested by - Heenan and Gilbert.
Compression in Inches of Water.
Efficiency Per Cent
Cubic Feet of Air per Minute
Efficiency at 2700 R.P.M.
Compression at 2700 R.P.M.
2 4 6 8 10 12 14
10 20 30 40 50 60 70
1000 2000 3000 4000 5000 6000
EUGENE DIETZGEN CO., CHICAGO.

From this data and the curves, the exit velocity polygons were drawn for various discharges and the theoretical rim speed computed by the method already described in detail. The results of these computations are given in the following table.

Capacity	Static Head	Radial Velocity	ϕ	Calculated Velocity	Rim Speed	Eff.
4000	7	22.4	90°	176	200	50
3500	9	19.6	"	181	"	60
3000	10.1	16.8	"	182	"	67
2500	11	14	"	188	"	69
2000	11.5	11.2	"	196	"	67

In the development of the theory, it was shown that with radial vanes, the fundamental equation was simplified as one of the factors reduced to unity.

$$u_2 = \sqrt{g\varepsilon\frac{h}{\eta}(1+\tan\beta\cot\phi)}$$

For radial vanes $\phi = 90^\circ$ this becomes

$$u_2 = \sqrt{g\varepsilon\frac{h}{\eta}}$$

This equation was used for computing the above table.

The discussion of the results of Mr. Guys tests upon his high pressure wheel applies to some extent here. Slipping undoubtedly occurs at the maximum discharge. Unlike the other results, however, there is still considerable difference between the actual and theoretical when the discharge has been throttled. This can only be accounted for by the unusually high efficiencies. The quantity of air in this experiment was measured by an anemometer which very often exaggerates the discharge. The natural result would be to augment the efficiency. It is very questionable whether an efficiency of 69% based on static pressure was

ever obtained with a wheel of this kind. Reducing the efficiencies to normal would make the comparison between actual and theoretical velocities very similar to those obtained for wheel No. 1 of Mr. Guy's experiments.

Some of the most valuable experiments upon small fans are due to Bryan Donkin. Unfortunately, however, the data is not in such form as to make it all available for a good comparison of the actual results and the theory previously presented. However, in one case, a fan with radial blades, the agreement is almost perfect. This particular machine was designated as fan No. 3. When running at 1248 R.P.M. it delivered 1657 cubic feet of air per minute at a pressure of 4.25 inches of water, with an efficiency of 61%. The general form of the blade is shown in figure 17.

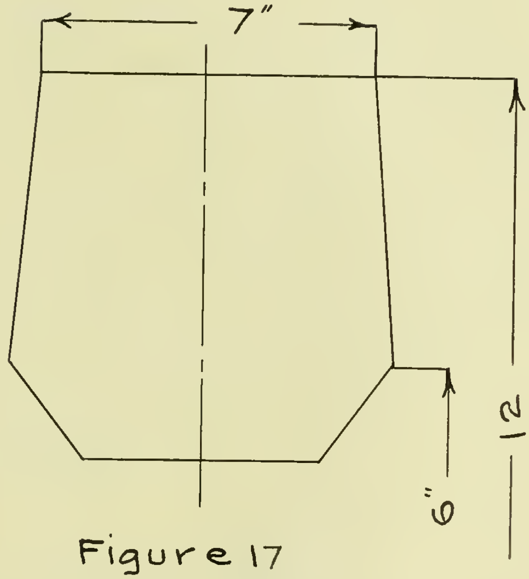


Figure 17

$$\frac{h}{\eta} = \frac{4.25}{.61} = 6.96$$

$$U_2 = \sqrt{g\varepsilon\frac{h}{\eta}} \qquad \text{(Radial Vanes)}$$

$$= \sqrt{32.2 \times 68.3 \times 6.96}$$

$$= 124 \text{ ft/sec.}$$

The actual peripheral velocity was $= \frac{24\ \frac{11}{}\ \times 1248}{12 \times 60} = 130$ ft./sec

In these experiments, the air was measured by means of the pitot tube and according to the judgment of various authorities, who have reviewed the results, the discharge as given may be relied upon. In view of these facts, the agreement here obtained between theory and experiment has particular significance.

Thus far, we have been concerned with the application of equations giving the relation between speed and pressure. We now come to another equation which is much more difficult to prove, namely that the mechanical efficiency

$$n = \frac{gH}{u_2w_2}$$

This is in many cases due to an exaggerated discharge which makes the experimental efficiency larger than it should be. Often it is difficult to determine the angle of discharge and consequently the value w_2. This depends upon various factors, such as contraction of the stream as it issues from the blast area of the wheel, or slipping of air over the ends of the blades. All of these influences distort the theoretical velocity polygons and make it extremely difficult to make an analytical analysis, which will be completely borne out in practice.

In deriving the equation

$$n = \frac{gH}{u_2w_2}$$

no account was taken of the work lost in bearing friction or in losses due to the rotation of the wheel disks in the air contained in the casing. This frictional effect varies as the square, and the lost work as the cube. It is evident therefore that at small orifices these losses become a larger and larger percentage of the total, and cannot be neglected if a close agreement is expected.

The equation, as it stands, can not be applied in the limiting case, namely shut-off. Here $w_2 = u_2$, and the mechanical efficiency becomes equal to manometric efficiency.

$$n = \frac{gH}{u_2 w_2} = \frac{gH}{u_2^2}$$

At shut-off, the manometric efficiency may be equal to one-half, while the mechanical efficiency must be equal to zero. From this it is clear that at small orifices and at shut-off, the expression $\frac{gH}{u_2 w_2}$ must differ considerably from the actual mechanical efficiency. We should expect the calculated value to be higher than the actual. When the blower is operating at the orifice for which it was designed, we should expect a closer agreement. However, with the many uncertainties which may enter into the calculations of the value of w_2, a close agreement cannot be anticipated.

To compare the above equation for the mechanical efficiency with the results obtained by experiment, the impellers previously considered will be employed here. The results are all presented in tabular form. In each case they have been obtained by making actual layouts of the velocity polygons, and thus graphically finding the values of w_2.

For wheel No. 1 of Mr. Guy's tests, the following results are obtained:

Capacity	u_2	w_2	Static Head	H	$\frac{gH}{u_2 w_2}$	Eff.
8500	283"	331	20.2	1380	.474	.63
8000	"	328	21.2	1445	.50	.64
7000	"	322	22	1500	.53	.64
6000	"	317	21.3	1450	.52	.60
5000	"	312	20.6	1401	.51	.53

Capacity	u_2	w_2	Statid Head	H	$\frac{gH}{u_2w_2}$	Eff.
4000	283"	306	19.5	1330	.495	.45
3000	"	300	18	1230	.465	.33

It is at once evident that on previous expectation in regard to the comparison at small orifices is fully justified. The values of w_2 given in the table were found by taking the direction in the same as the vane angle. As previously shown, the vanes of this impeller are so constructed that the angle of discharge in all probability is greater than the theoretical. This would decrease the actual value of w_2 and increase the calculated value of the efficiency, making the agreement closer than that shown in the table.

Wheel No. 2 of Mr. Guy's experiments gave the following results:

Capacity	u_2	w_2	Static Head	H	$\frac{gH}{u_2w_2}$	Eff.
5750	283	53	4.3	294	63.2	53
5250	"	73	5	342	53.5	58
4500	"	103	6.3	430	47.5	56
3500	"	142	7.8	533	44.4	49
2500	"	183	9	615	38.2	38
1500	"	223	10	683	35	25

With the impeller the agreement at the rated discharge, 5250 cubic feet per minute, is as good as can be expected. Although this wheel has only six vanes, the passages are so shaped that the air is properly guided and the general direction of the channel between any two blades should conform very closely to the direction of the blades themselves.

Wheel No. 1 of Heenan and Gilbert's experiments had an efficiency of 52% with a delivery of 2500 cubic feet per minute, and a compression of 6.8 inches water gage.

Here $u_2 = 200$ ft./sec

$w_2 = 180$ ft./sec (From velocity polygon)

$$\text{Eff.} = \frac{32.2 \times 6.8 \times 68.3}{200 \times 180} = 42\%$$

For this fan, there is a rather large divergance between the actual and calculated efficiency.

When an impeller has vanes with radial tips, the velocity components w_2 becomes equal to the rim velocity u_2. Therefore

$$\text{Eff.} = \frac{gH}{u_2 w_2} = \frac{gH}{u_2^2}$$

The last expression is also by definition the pressure or manometric efficiency. Hence for this form of fan the mechanical and manometric efficiencies should be equal.

For wheel No. 2, constructed with vanes having radial ends, the following tabular results are obtained:

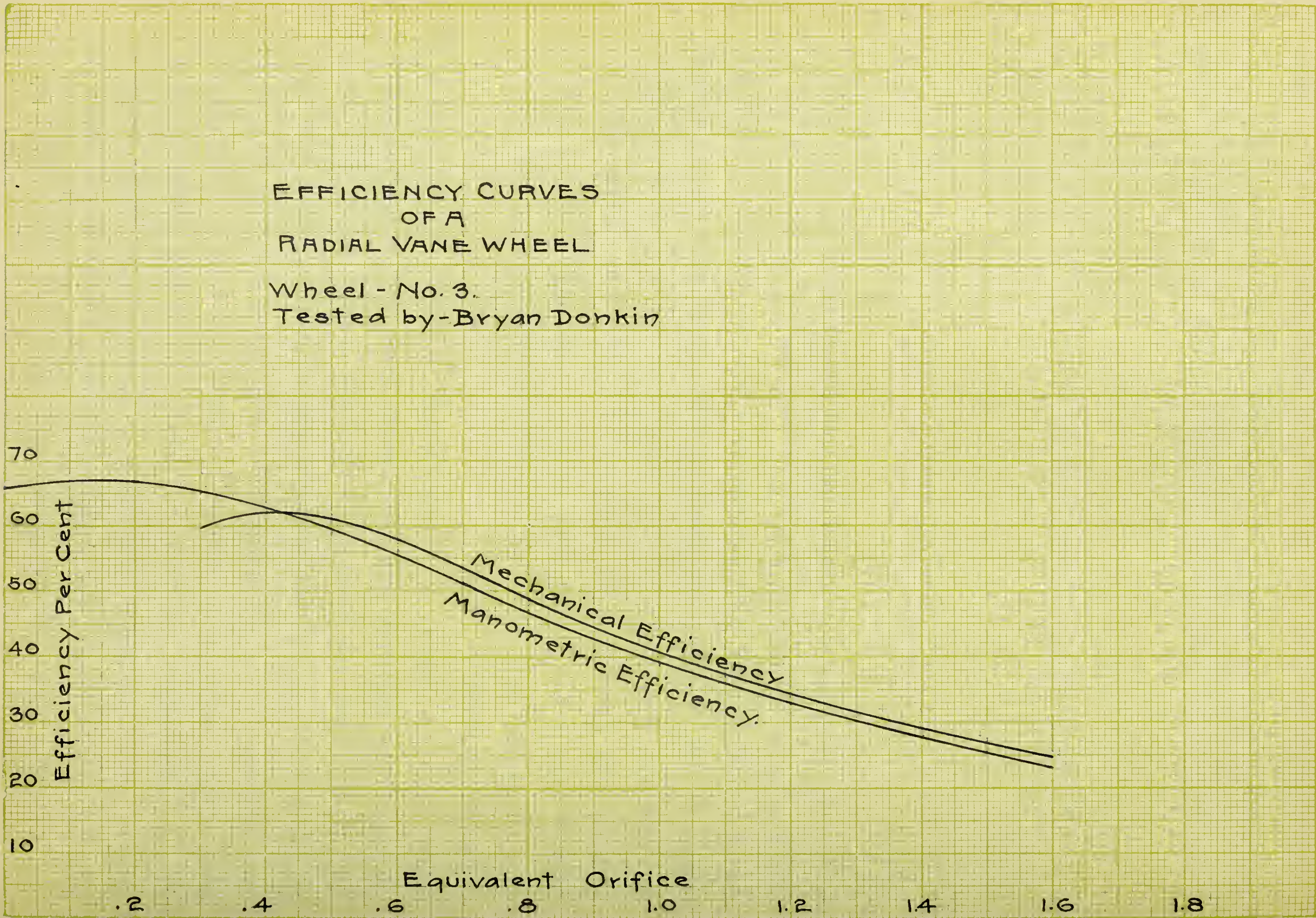
Capacity	u_2	Static Head	H	$\frac{gH}{u_2^2}$	Eff.
4000	200	7	480	38.4	50
3500	"	9	615	49.5	60
3000	"	10.1	690	55.5	67
2500	"	11	752	60.5	69
2000	"	11.5	786	63.2	67

Even at the smallest orifice, the calculated efficiency is less than the value obtained by experiment. This is contrary to the results that the theory presented would indicate. In the first analysis of this test attention was called to the probable exaggeration of the discharge and corresponding mechanical efficiency. The above comparison confirms this statement.

The most complete verification of the law $n = \frac{gH}{u_2 w_2}$ is obtained from the results of Bryan Donkin's experiments. Fan No. 3 with radial blades has already been found to conform very closely to the relation laid down by the fundamental equations between circumferential velocity, and compression. With impellers of this type the mechanical and pressure efficiencies should be equal. Both of these curves are included in the report of the tests. They have been replotted to the same scale, and are shown upon page 73 . The agreement is remarkable; the difference is never more than $2\frac{1}{2}\%$ except at small orifices where a difference should occur.

What conclusions may be drawn from the foregoing analysis and discussion of blower tests? This question is best answered by a brief resume of the scope and results of the preceeding discussion. Attention is called to the wide range of conditions to which the theoretical equations have been applied. Some of the impellers were designed for high pressure, others for low pressure, and still others for intermediate pressure service. Some for high, and others for medium rotative speed. Various forms of vanes were represented; radial, inclined forward, and inclined backward.

The fundamental equation for peripheral velocity which , though not showing exact agreement, gave results which in nearly all instances can be considered reasonably close. In some cases, the agreement was remarkable; in a few the divergance so large as to cast some doubt upon the applicability of the equations. However, when the conditions were closely analyzed, sufficient reason was always found to account for, in a measure at least, the seeming



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discrepancy. The largest differences always occured when some of the primary assumptions of the theoretical analysis had been violated. It is extremely difficult to determine the exact angle of the discharging air which is nearly always different from the vane angle. This alone usually accounts for most of the difference between theory and experiment, when the experimental results are reliable. We are consequently led to the following conclusion, namely that the preceeding analysis clearly shows that experiment does coincide with the theory developed by Weisbach and Innes, when the latter is intelligently applied, and all conditions properly observed.

The equation for mechanical efficiency, however, must be applied with considerably more caution. The reason is obvious, not only the design of the impeller, but also the volute and bearings all have their own influence upon the mechanical efficiency of the machine. It would be manifestly impossible to take all of these factors into account in a simple equation. However, this method of predicting the efficiency may be employed as a guide when attempting to establish the performance of a new machine.

A study of the characteristic curves of a steam turbine shows at once that a high rotative speed is essential to good economy. The difficulty which has been encountered in designing turbine driven units has mainly been in devising machines which would allow the turbine to operate at its economical speed. The advent of the turbine caused an evolution in the design and construction of centrifugal pumps, blowers, and generators. Very often it has been preferable to allow the driven machine to operate with a smaller medhanical efficiency if the rotative speed could thereby be increased.

The loss thus incurred in the driven machine is more than offset by the gain in the turbine, and consequently in the overall efficiency of the unit.

One of the most difficult machines for applying turbine drive is the volume blower. With low pressures and large volumes the rim velocity is limited on the one hand, and the reduction in diameter of the wheel on the other. With a practically fixed peripheral velocity, the first step would naturally be to choose an impeller of small diameter in order to secure a high rotative speed. The entrance velocity which may be employed for efficient performance is much lower for low pressure than for high pressure blowers. This velocity determines the inner diameter of the wheel. To secure a proper guidance of the air within the wheel itself, the channels must have sufficient length. These conditions impose severe restrictions upon the diameter of the rotor. Blowers have been constructed in which an attempt has been made to raise the entrance velocity. In order that the air will flow into the impeller a difference of pres-

sure must exist which will vary with the square of the velocity. As the pressure without is constant, a slight vacuum must be created within the wheel. If the entrance velocity has been chosen too high the action of the blower will not cause sufficient depression to maintain the required velocity. The result is that the flow ceases to be uniform and cavitation takes place. At the present time, there are comparatively few direct connected turbine driven volume blowers in service. The general practice has been to employ geared units where turbine drive was desirable. It is impossible to design a fan which will compress air at high speeds as efficiently as one operating at low speeds. The entrance, windage, surface friction, and eddy losses are all increased. For these reasons, we cannot expect a blower to give the same mechanical efficiency when operating at turbine speeds as one operating directly connected to a reciprocating engine. The problem is rather to design a fan which will develop a fair efficiency, and which can be run at an economical turbine speed. The speed of the average volume blower is between 200 and 300 revolutions per minute. This excludes them from all consideration for a direct connected unit.

The first step in designing a turbo-volume blower is to choose the vane angles so as to give the maximum circumferential velocity for a given compression. The fundamental equation giving the relation between these factors is

$$u_2 = \sqrt{g\varepsilon(h+z)+\frac{w^2}{2})(1+\tan\beta\cot\phi)} \qquad \text{———} \quad \text{———} \quad \text{———} \; 8$$

Let $\sqrt{1+\tan\beta\cot\phi} = K$

Then $u_2 = K\sqrt{g\varepsilon(h+z+\frac{w^2}{2})}$

Obviously the angles should be so chosen to make K as large as possible. When $\phi < 90^\circ$ the cotangent is positive. As the cotangent increases very rapidly in value as it approaches zero, K can be made large by taking the value of ϕ as small as possible. In other words the tips of the vanes should be bent back as far as practicable. This construction was employed by Mr. Albert Guy in Wheel # 2, which was operated at 3600 R.P. M. for a compression of five inches of water. In this wheel, the angle ϕ was equal to 10°. This is past the practical limit for the reason that the air passages through the impeller become very long, thus increasing surface friction, and because it becomes difficult to connect the entrance and exit tips of the vanes by means of a smooth curve which will also give the proper channel cross-section, especially when the wheel is to be designed with a short radial depth.

The influence of the discharge angle β upon the peripheral velocity is not so marked. It is evident, however, from equation 8 that for values of $\phi < 90^\circ$ u_2 will be increased by increasing β. It would of course necessitate a high radial velocity from the wheel. Attempts to secure high speed in this way have failed. Tests upon blowers have shown that the discharge angle cannot increase beyond a more or less fixed value. The difficulty seems to be due to the radial velocity. If the value for this has been taken too large, the blower will not operate efficiently at the capacity for which it has been designed. The peak of the efficiency curve will occur before the desired delivery has been attained. In order to be certain that the fan will deliver the volume of air required the discharge angle should not be taken too large. No arbitrary angle can be given for the largest value of β. In a few tests thirty degrees has given fair results but this can be re-

garded as approaching the upper limit. In a well conducted turbo-blower test the best efficiency at all speeds was obtained at that head and capacity where the discharge angle was equal to 25 degrees.

There is another reason why it is desirable to choose β large in addition to its effect upon the rim velocity. If several velocity diagrams are constructed with different values of β all for the same compression it will be noted at once that as the discharge angle is increased the absolute velocity at exit becomes smaller. This signifies that a greater pressure change has taken place in the impeller, and that a smaller amount of the total compression is obtained by transforming the kinetic energy due to the velocity of the air leaving the impeller into pressure energy. This transformation, which takes place in the diffuser and volute, is usually accompanied by heavy losses. It is therefore desirable to obtain as much change of pressure as practical within the wheel itself.

As soon as the vane and discharge angles have been determined the peripheral velocity u_2 is fixed. The rotative speed depends then solely upon the diameter of the wheel. At this point care must be exercised so that the outer diameter will not be taken too small. To obtain good performance, the blades must have sufficient radial depth. This consideration partly determines the inner diameter of the wheel. Attention has already been directed to the results which obtain when the inlet area has been unduly restricted. It is therefore impossible to make the outer diameter any smaller than necessary to secure sufficient inlet area in the first place, and sufficient radial depth of blade in the second. Mr. Albert Guy obtained good results with an inlet velocity of 58

feet per second, which is considerably above the normal. Even with this value the size of wheel will be larger than desirable for good turbine speeds, especially, with blowers of large volume, such as are required for forced draft service.

A method will now be outlined by means of which the difficulties due to a large entrance velocity can be overcome. This consists essentially of providing mechanical means for supplying air to the entrance tips of the vanes. When this is done cavitation cannot take place, because the flow of air into the wheel is no longer dependant upon the depression set up in the entrance passages due to the discharge of air from the periphery of the impeller. The simplest means to induce the flow of air is the propeller fan. It is perfectly feasible to set a propeller wheel in each opening of the fan rotor so designed to supply the rated volume of air when the machine is operating at the proper speed.

It is impossible however to construct a propeller wheel which will deliver air with a purely axial flow. In passing through the wheel, the air will receive a tangential component which causes the air to be discharged with a whirling action. Radial inflow to the wheel would, under these circumstances, be impossible. One of the conditions for maximum efficiency is that entrance should take place without shock and by the shortest path. The shortest path is obtained when inflow takes place radially. This condition, however, cannot be observed if a propeller wheel is employed for bringing the air into the wheel. In addition, it would be very difficult to shape the blades so that inflow would take place without shock. As the whirling motion of the air is retarded as it progressed through the wheel, the pick-up angle would take different values across the wheel width to conform to the change of tangential

velocity of the entering air.

These considerations render the direct application of the propeller unsatisfactory for the purpose proposed. There is an expedient, however, by means of which the preceeding difficulties can be overcome. By placing a set of guide vanes in front of the propeller wheel, which is designed to receive air in an axial direction and discharge it with a tangential component, the propeller can be made to deliver air into the wheel in a purely axial direction. With this arrangement it is essential of course that the moving vanes are so shaped that the inflow will take place without shock, when the blower is operating at its normal orifice.

The arrangement proposed here is open to several objections. The surface friction of the machine is largely increased by the introduction of special moving and stationary vanes. Inflow without shock can take place only at one delivery; for all other capacities there will be some loss of head due to the impulse imparted to the air in entering the propeller wheel. The natural consequence of this condition is to cause the mechanical efficiency curve to drop more rapidly on both sides of its maximum value. In other words, it will be difficult to obtain a flat efficiency curve.

The objections raised here are both valid, but are not sufficient to exclude the proposed arrangement. By these means, it is possible to operate with a smaller wheel and higher rotative speed. The gain in economy of the turbine resulting from the possible increase of speed will more than off-set the losses incurred in the fan. Furthermore, we are enabled to run at turbine speed, which is impossible with the standard type of wheel.

With blowers designed for high inlet velocities, careful attention must be paid to the construction of the inlets. All unnece-

essary obstructions to the free passage of the air must be avoided. If these precautions are not observed, considerable loss of head will result at inflow, and the required delivery will not be obtained. To reduce the entrance loss to a minimum, a bell-shaped mouth piece should be fitted to the casing. The bearing supports should be designed so as to offer the least resistance to the entering air.

The effect of the bell entrance nozzle was shown by the experiments of Bryan Donkin. Two tests were made upon a Rateau Fan which was designed with a bell-mouth inlet. In the first test, this was in place;in the second test, it was removed. The result showed that at the same speed 31 percent more air was passed when the inlet was used and the mechanical efficiency was 9 percent higher. This clearly proves the advantage of admitting the air without shock, and in the proper direction.

The function of the casing is two-fold. It not only forms the volute which receives the air discharged from the impeller and guides it to the mouth of the fan, but it also serves as a means of support for the bearings. To fulfill its purpose as a bearing support, it must have proper stiffness. In designing a casing due allowance must be made for the lateral pressure which is set up within the volute. For example, let the static pressure at the fan outlet be equal to five inches of water. Now an inch of water is equal approximately to a pressure of five pounds per square foot. Thus, in this case, the total lateral pressure would be about twenty-five pounds per square foot. In large blowers, in which the casing has a considerable area, ample means must be provided for taking care of this pressure.

The current practice of blower manufacturers is to use sheet

steel for the casing, and provide angle iron or channel stiffness for supporting the sides. A much more elegant construction consists of cast iron sides with a sheet iron back. This not only completes the casing, but holds the sides firmly in place. With cast iron sides, supports for the inlet piece and bearing bracket can easily be provided. A blower built in this way can never warp out of alignment. This construction insures good running conditions, which are essential for a turbine driven unit.

The requirements for the bearings of a turbo blower are well brought out in a set of specifications recently issued by the mechanical engineer of the New York Edison Company. These specifications were drawn up to serve as a basis for the proposals upon some turbo blowers for supplying air for forced draft. The bearings were to be babbit lined, ring oiling, and oil tight.

For high speed service, a ring oiling bearing is absolutely essential. The function of the rings is to flood the journal with oil. In order to accomplish this purpose, rings of special construction must be used. The common practice of employing a small ring made of round steel should be abandoned. One of the best methods for raising oil from the bearing reservoir is due to the Burke Electric Co. This consists of a ring made up of a number of elements. These elements are punched from zinc plate. This ring is especially adapted for light oil.

The oil reservoir should be generously proportioned; not only is provision thus made for a large oil supply, but there is a much better opportunity for the radiation of heat due to bearing friction, which has been absorbed by the lubricating oil. There is no established practice in regard to oil grooves. When properly constructed they are unquestionably an aid to good lubrication. Burke rings,

however, will carry considerably more oil to the shaft than will ordinarily be wiped in. It is good practice, therefore, to provide a well in the bearing shell on the retreating side of the ring. This will catch a large amount of the oil, and thus provide a means for aiding its flow into the bearing.

The bearing shells should be split horizontally, and firmly held to-gether with dowel pins. To obtain the best results with babbit lined bearings, the lining must be securely held by proper anchors. The best form of anchors are dove tail grooves cored into the bearing shell. Another mechanical feature that should be observed in designing a bearing of this type is to make provision for removing the bearing shells without disturbing the shaft.

All operators of blowers have experienced a great deal of difficulty in keeping the oil within the bearing case. It is for this reason that oil tight bearings are demanded. The large depression due to the high inlet velocity will draw the oil from the bearing if there is the slightest possibility of leakage. Some makers use leather collars set into the bearing case to make them dust proof and to prevent leakage. This form of construction has not proven entirely satisfactory. A method has recently been proposed which will effectively take care of this difficulty. A simple stuffing box, containing several rings of packing, is provided on each side of the bearing case. This application of the stuffing box has proven successful in practice and will undoubtedly be adopted as standard of good construction.

The essential requirements of a turbo-blower have now been outlined. Before passing to the design, however, it may be well to give a brief summary of the advantages of the proposed construction. The curvature of the vanes will permit a high peripheral

velocity. The propeller wheel will introduce the air axially into the wheel, so that it may flow into the vanes without shock. This not only reduces the entrance loss, but also permits the employment of a high entrance velocity without danger of cavitation. By this expedient, a wheel of smaller diameter and consequently greater rotative speed can be used. To secure good entrance conditions, a properly shaped inlet piece is provided. The cast iron casing sides will make a rigid support for the bearings. The bearings are ring oiling and oil tight designed not only to be accessible, but to give the highest mechanical efficiency. This construction has been proposed not only to fulfill all operating conditions imposed upon such a blower, but also to permit the use of steam turbine drive, and make the unit smooth running, and give reliable and lasting service.

VII -- DESIGN OF A TURBO VOLUME BLOWER.

The problem proposed in the beginning of this thesis was to design a turbine driven blower which should supply air to Taylor stokers, and when installed, should fulfill existing operating conditions; supplying the proper volume of air at the stoker tuyeres with the required pressure. An analysis of conditions imposed upon this blower showed that an efficient operation of the boilers will be secured when 30,000 cubic feet of air per minute are delivered to the stokers. Maintaining a static pressure of five inches of water in the wind-box necessitated a corresponding pressure of six inches of water measured at the discharge opening of the blower.

The underlying theory governing the design of centrifugal fans has been presented to-gether with an analysis of experimental results which clearly indicated the extent which the fundamental equations govern actual performance. In addition, the mechanical constructional requirements essential to reliable operation have been outlined, and the general type of blower selected. The theory may now be applied by carrying through the complete design of a blower, which is to meet the requirements previously stated.

The first step in the design of a machine of this type is the determination of the inner radius of the impeller. This dimension is governed by the inlet velocity. On account of the mechanical means adopted for insuring a steady flow of air into the wheel, a high inlet velocity may be chosen. With this arrangement 100 feet per second may be assumed for a trial value.

It is not feasible to carry the blades of a propeller wheel to a small hub. In the first place, sufficient blading at the rim will cause a congestion at the hub and difficulty will be encountered in fastening the blades. In the second place, it becomes impossible

to give the blades the proper form close to the center of the shaft on account of the wide difference in linear velocity. Both of these difficulties are obviated by using a hub of larger diameter. If this is taken as approximately one third of the diameter of the wheel, good results are obtained.

Take entrance velocity equal to 100 feet per second.

$\frac{30,000}{60}$ = 500 cubic feet per second

$\frac{500}{100}$ = 5 square feet of net entrance area.

As the impeller will be of the double suction type, 2.5 square feet, or 360 square inches of entrance area will be required on each side.

Take 24 inch circle

Area 24 inch circle	=	452.39 sq. in.
Area of 8 inch hub	=	50.26 " "
Net area		402.13 sq. in.

Inlet velocity $= \frac{500 \times 144}{2 \times 402} =$ 89.5 feet per second.

The assumed value may safely be taken for the inner diameter of the wheel.

To find the width of the impeller, the inlet area is first assumed equal to the entrance area.

Let X = half the width

Then $24 \pi X = 402$

$$X = \frac{402}{24 \pi} = 5.35 \text{ inches}$$

Take the width equal to six inches, or twelve inches for the complete impeller.

Inlet area to the vanes $= 24 \times \pi \times 12 = 905$ sq. in.

Inlet velocity to the vanes $= \frac{500 \times 144}{905} = 79.5$ ft. per second

To obtain a high rotative speed, the outer diameter must be

made as small as possible. A radial depth of six inches is sufficient for proper form of channels between vanes. The outer diameter is thus fixed at 36 inches. On account of the comparatively short radial depth of blade, the width of the wheel will be made constant. This fixes the radial velocity at the rim of the wheel.

Discharge or blast area $= \pi\ 36 \times 12 = 1360$ sq. inches

Radial velocity $= \dfrac{500 \times 144}{1360} = 53$ feet per second

Adding 3% for thickness of vanes, we obtain 55 feet per second as the actual velocity of discharge in a radial direction from the wheel.

To secure good efficiency, it will be necessary to choose such a combination of vane and discharge angle that not only will give the desired compression, but will also give the radial velocity already determined. This can only be done by a series of trials, choosing different angles until the proper radial velocity has been obtained.

In making the analysis of blower tests, the fundamental equation for the peripheral velocity of the impeller was transformed from the standard form

$$u_2 = \sqrt{g\varepsilon(h+z) + \frac{w^2}{2})(1 + \tan\beta \cot\phi)} \qquad \text{——— ——— ———} \quad 8$$

to $$u_2 = \sqrt{g\varepsilon \frac{h}{\eta}}\ \sqrt{1 + \tan\beta \cot\phi}$$

Here n is the efficiency based upon static pressure, and is equal to $\dfrac{h}{h+z+\frac{w^2}{2g\varepsilon}}$ Before the value of u_2 can be calculated, the efficiency of the machine should be known. At this point, considerable experience and good judgment are necessary. If the anticipated efficiency is not realized, the required compression will not be

obtained. If, on the other hand, the assumed value is too small the blower will have to operate under the specified speed. In a steam turbine, the horse power drops rapidly with a decrease in speed so that the power output of the turbine may be insufficient to drive the blower. In addition, the angles of the vanes are laid out for some given speed and if this does not obtain, the maximum efficiency cannot be obtained at the rated pressure and delivery.

The efficiency of this blower will be based upon total pressure. For this type, of blower, build with good workmanship in strict accordance with this design, an efficiency of 62% should be obtained.

$$\frac{h + \frac{w^2}{2g}}{h + \frac{w^2}{2g} + z} = .62$$

The duct velocity is 70 feet per second.

Substituting in the above equation, we obtain

$$\frac{6 + \frac{\overline{70}^2}{2 \times 32.2 \times 68.3}}{6 + \frac{\overline{70}^2}{2 \times 32.2 \times 68.3} + z} = .62$$

$$\frac{6 + 1.1}{6 + 1.1 + z} = .62$$

Solving for z

$$z = \frac{7.1}{.62} - 7.1$$

$$= 11.45 - 7.1$$

$$= 4.35$$

Then

$$\eta = \frac{h}{h+z+\frac{w^2}{2gE}} = \frac{6}{6+4.35+1.1} = .52$$

Substituting in the fundamental equation

$$u_2 = \sqrt{68.3 \times 32.2 \times \frac{6}{.52}} \sqrt{1+\tan\beta\cot\phi}$$

$$= 159\sqrt{1+\tan\beta\cot\phi}$$

Let

$$\phi = 20^\circ \quad \text{and} \quad \beta = 28^\circ$$

Then

$$\sqrt{1+\tan\beta\cot\phi} = \sqrt{1+.5317\times 2.747} = 1.57$$

Then

$$u_2 = 159 \times 1.57 = 250\,\text{ft}/\text{sec}.$$

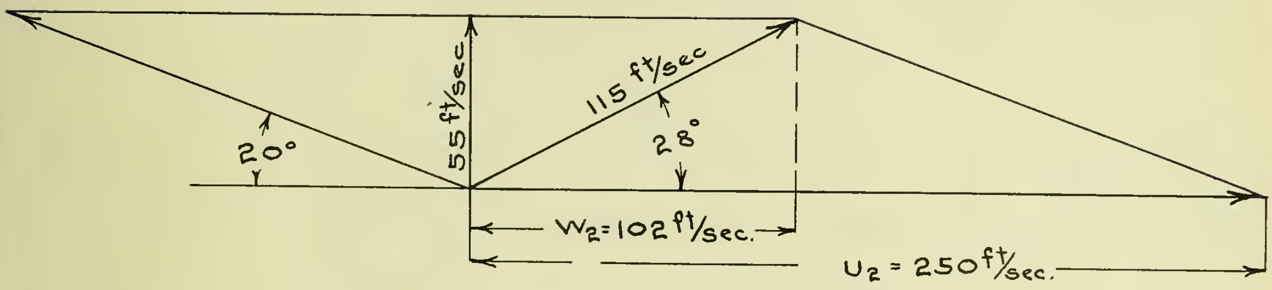
The velocity polygon for exit may now be drawn and is shown in figure 18 . The radial velocity scaled from the diagram is 55 feet per second. This is the required value, so that the assumed values of β and ϕ are correct.

Before fixing the rotative speed, an allowance must be made for slipping over the tips of the vanes, as experiments show that such an action always takes place.

$$\frac{263}{.925} = 270 \text{ feet per second}$$

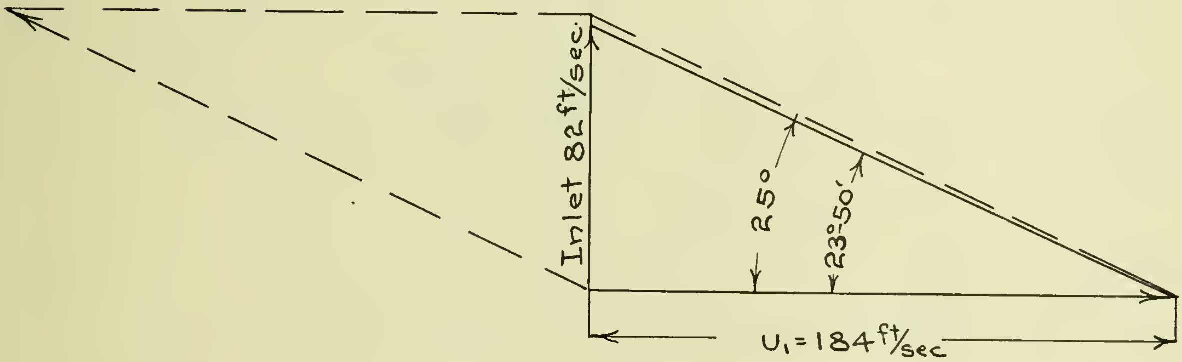
$$\frac{270 \times 60}{3 \times 3.14} = 1720 \text{ r. p. m.}$$

For operation, take speed as 1750 r.p.m.



Velocity Polygon at Exit

FIGURE 18



Velocity Polygon at Entrance

FIGURE 19

The useful work has been found to be

$$\frac{Qhy}{33000}$$

Here h is the static pressure in feet of air. Substituting, we have

$$\text{Air Horse Power} = \frac{30,000 \times 6 \times 68.3 \times .076}{33000} = 28.4$$

With an efficiency of 52% based on static pressure

$$\text{Turbine Horse Power} = \frac{28.4}{.52} = 54.6$$

Referring to the calibration curve of the form 187M Kerr Turbine, we note that the required horse power is obtained at 1750 R.P.M. with a steam pressure of 105 pounds per square inch upon the first stage, or about 120 pounds on the main. This difference of pressure is more than ample for governing.

Applying Inne's equation for efficiency, we have from the velocity diagram

u_2 = 250 feet per second

w_2 = 102 feet per second

$$\text{Then } \frac{gH}{u_2 w_2} = \frac{32.2 \times 6 \times 68.3}{250 \times 102} = 51.8\%$$

This value coincides very closely with the assumed value 52%.

The inlet velocity to the vanes was found to be 79.5 feet per second. Adding 3% for thickness of vanes, this is increased to 82 feet per second. For entrance without shock, we have the condition

$$\cot = \frac{u_1}{c_1}$$

$$u_1 = \frac{1750 \times 24}{229} = 184 \text{ ft/sec}$$

$$c_1 = 82 \text{ ft/sec}$$

$$\cot\theta = \frac{184}{82} = 2.25$$

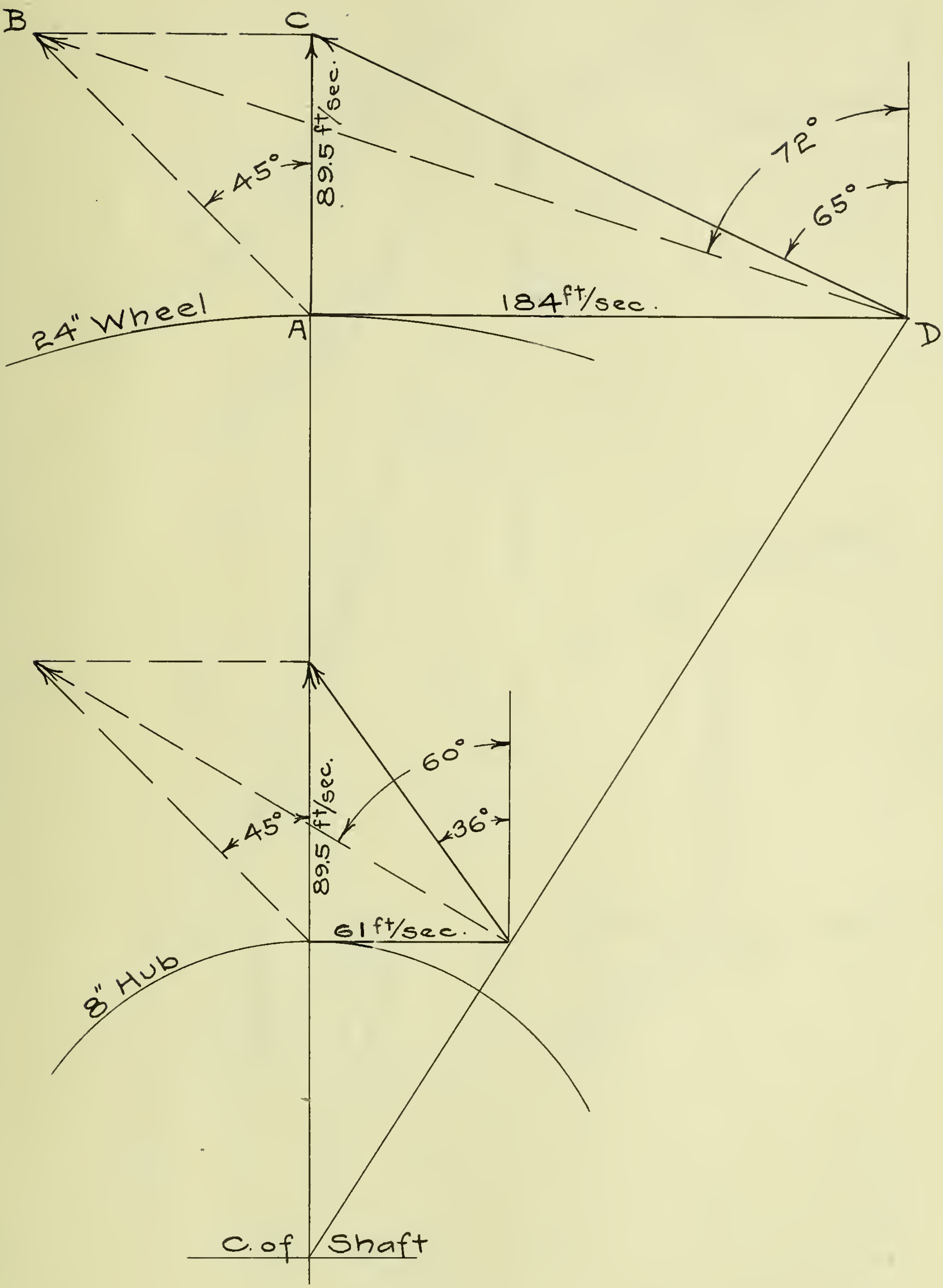
$$\theta = 23^\circ - 50'.$$

In order that the required capacity will be obtained at maximum efficiency, it is best to make the pick-up angle slightly larger than the theoretical value. The velocity diagram at entrance is shown in figure 19 with both theoretical and actual angles; The latter being one degree larger than the former.

The shape of the guide vanes, which direct the air into the propeller, is governed by two conditions. They must receive the air axially and discharge it with a tangential component opposite to the direction of rotation. The direction of discharge is not fixed, but should be so chosen that the pick-up edge of the propeller wheel itself will take a desirable form. After several trials, an angle of 45° was selected.

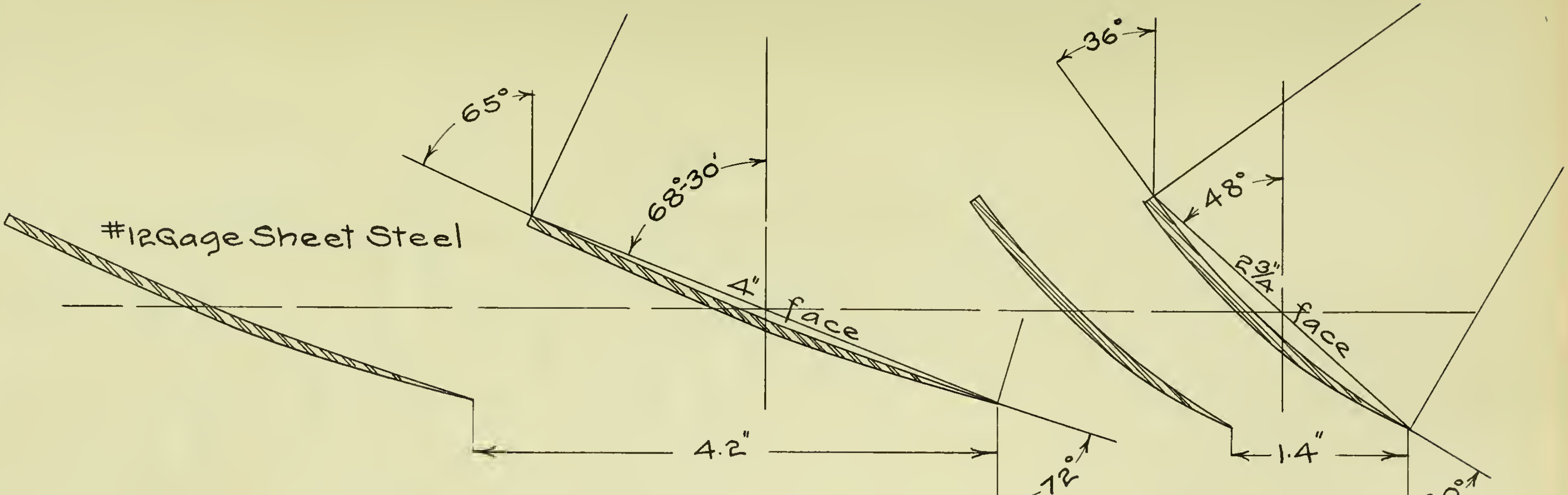
A section taken at the rim, and another at the hub is shown on page 94. On account of the almost axial direction of the blades a larger number was selected than chosen for the propeller wheel, in order that proper guidance may be secured. The guide vanes are made of No. 12 sheet steel and riveted to steel rings at the inner and outer circumferences. The assembled vanes are then set into the casing and secured with screws, before the bearing heads are put into place. The general detail of construction is shown in the assembly view of the blower on page 103.

The blades of the propeller wheel must be so formed that the air is received without shock from hub to rim, and discharged in a purely axial direction over the entire area. The velocity polygons which determine the curvature of the blades are shown on page 93. The entrance diagram is in dotted lines to distinguish it from the exit diagram, which is in full lines. As shown, the air enters



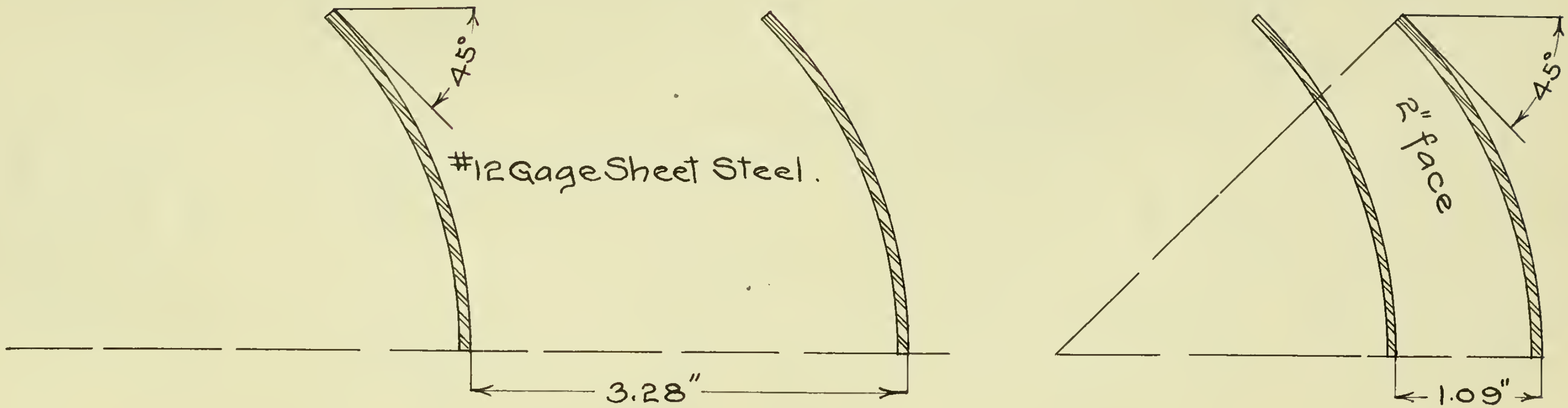
Velocity Diagrams
for Propeller Wheel.

Propeller Blades



Rim Section 18 blades

Hub Section



Rim Section 23 blades

Hub Section 23 blades

Guide Vanes

with an inclination of 45° away from the direction of rotation. The axial component of the inlet velocity has already been found to be 89.5 feet per second. This component combined with the determined inclination, gives the absolute velocity of entrance AB. The velocity AB combined with the rim velocity AD determines the relative velocity of entrance DB. The entrance angle of the propeller blade is equal to the inclination of DB.

The absolute velocity of exit is AC. When combined with the peripheral velocity AD, the relative velocity of exit DC is determined. This velocity in turn fixes the angle of the discharging edge of the blade. The velocity polygons at the hub are constructed in a similar manner. It will be observed at once that on account of the varying peripheral velocity from hub to rim, the angles of entrance and discharge are not constant. If the propeller blade is long as many points can be chosen as are deemed necessary to define the proper shape of the entire blade.

From the angles thus fixed by the velocity diagrams at entrance and exit, the section of the blade can be laid out. The construction is shown in detail on page 94. The number of blades can best be found by making several layouts and choosing that number which gives the best spacing. For this blower 18 blades were taken, even though the rim sections are rather far apart. The hub sections, however, are as close to-gether as they can be placed and fastened.

The reason for choosing twenty-three guide vanes can now be made clear. If the number of guide and propeller vanes are not incommensurable, the blower will operate with a siren action. The rotation of one wheel close to another one, which is fixed, will give a note, the pitch of which corresponds to the number of open-

ings multiplied by the revolutions per second. Allowing ample clearance between the fixed and moving vanes, in addition to choosing incomensurable numbers, will aid in securing silent operation.

The hub of the propeller is made of cast steel. The blades are inserted as shown in figure 10.

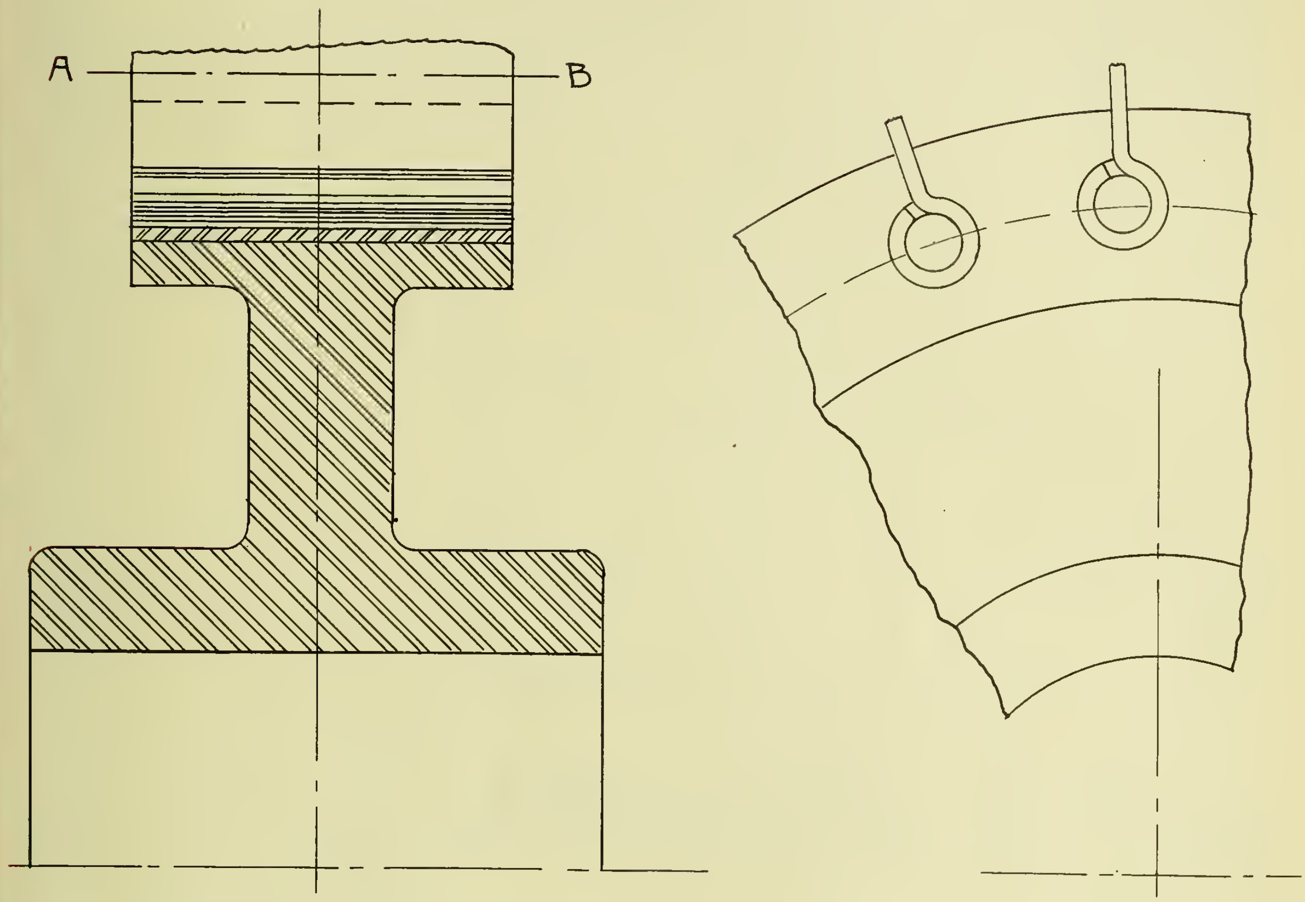


Figure 20.

An eye is turned on the end of the blade which is put into a slotted hole cut into the hub. The blade is then fastened in place by compressing a steel pin within the eye with a hydraulic press. When the blades are being fastened, it is necessary to rivet the pins

alternately around the rim so that the hub will not be distorted.

The centrifugal force due to the weight of the blade and its angular velocity is resisted by the tension in section AB. The stress in this section must be carefully computed so that the wheel can run at the required speed with an ample margin of safety. To find the centrifugal force acting, the weight of the blade and its center of gravity must be accurately determined. In figure 21 is shown the blank for a blade; these are made of No. 12 gage sheet steel, which weighs 4.375 lbs. per square foot.

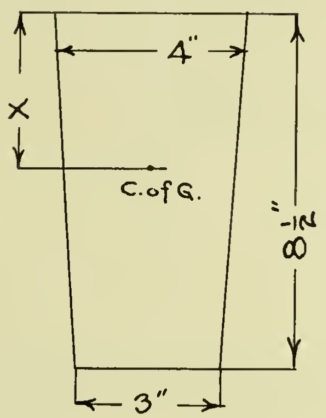


Figure 21

Weight of blade $= \frac{3.5 \times 8.5}{144} \times 4.375 = .905$ lbs.

Applying the equation for the coordinates of the center of gravity of a trapezoid, we have

$$X = \frac{4+6}{4+3} \times \frac{8.5}{3} = 4.05''$$

Radius of center of gravity is $12 - 4.05 = 7.95$ inches.

Centrifugal force $F = mrw^2$

$$F = \frac{.905}{32.2} \times \frac{7.95}{12} \times \left(\frac{2\,\pi\,1750}{60}\right)^2$$

$$= 625\ \#$$

Take width of hub as 2 inches. Then area of section AB is

$2 \times .109 = .218$

$\frac{625}{.218} = 2920^{\#}/\text{sq. in.}$ Tension at root of blade.

To calculate the elongation of a blade due to centrifugal tension the following equation has been deduced.

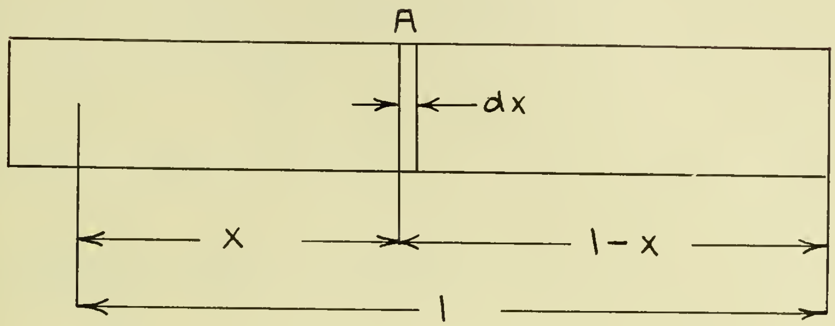


Figure 22.

In figure 22 let:

a = area of cross section

l = length

e = elongation

w = weight per cubic foot

ω = angular velocity

E = modulus of elasticity

Take any section A at radius x. The force exerted on this section is that due to portion (l - x).

Weight of section l - x = $(l - x)aw$

Radius of c. of g. $= x + \left(\frac{l-x}{2}\right) = \frac{l+x}{2}$

Centrifugal force $= \frac{aw}{g}(l-x)\left(\frac{l+x}{2}\right) \times \omega^2$

$= \frac{aw\omega^2}{2g}(l^2 - x^2)$

Fiber Stress at A $= \frac{w\omega^2}{2g}(l^2 - x^2)$.

Elongation of element dx is $\frac{\text{Fiber Stress}}{E}\,dx$

Substituting, we have

$$de = \frac{w\,\bar{w}^2}{2\,gE}\,(l^2 - x^2)dx$$

$$e = \frac{w\,\bar{w}^2}{2\,gE}\int_0^l (l^2 - x^2)dx$$

Integrating between limits l and 0.

$$e = \frac{w\,\bar{w}^2}{2\,gE}\,\left(l^3 - \frac{l^3}{3}\right)$$

or between limits l_2 and l_1

$$e = \frac{w\omega^2}{2gE}\left[l_2^3 - \frac{l_2^3}{3} - l_2 l_1 + \frac{l_1^3}{3}\right]$$

Applying this equation

l_2 = 1 foot

l_1 = .33 "

ω = 58.4 Π

w = 480 pounds

$$e = \frac{480 \times \overline{58.4\pi}^2}{64.4 \times 30{,}000{,}000 \times 144}\left[1 - \frac{1}{3} - .33 + \frac{.036}{3}\right]$$

= .0000183 feet

= .00022 inches

The ends of the propeller blades are riveted to an angle iron ring which is in turn riveted to the sides of the impeller. This form of construction not only stiffens the blades, but they in turn hold the runner in rigid balance. Considerable difficulty has been encountered, when operating squirrel cage runners at high speed. If the outer ring deflects out of balance, it will wreck the wheel. The usual method of staying this ring with tension rods, has proven ineffective in many instances.

The vanes of the impeller proper are made of No. 10 gage sheet steel flanged on both sides and curved to give the required entrance

and exit angles. One side of the vane is riveted to a heavy central disk and the other to a side plate of sheet iron. The central disk is supported and driven by a deflector keyed to the shaft. This deflector also serves to smoothly guide the air into the wheel in a radial path. The general construction of the completely assembled runner is shown in the drawing of the blower on page 103. Before assembling the runner and propeller wheels are balanced individually. After assembling, the rotor is again balanced. In this way, both static and kinetic balances are obtained and smooth running secured.

The blower shaft is subject to the tension developed by the shaft horse power of the turbine, and to bending due to the weight of the rotor. In finding the required diameter, the usual method of calculation leads to diameters which are too small for the high rotative speeds. In this blower, the size of the shaft in the bearings is taken equal to the diameter of the turbine shaft. Between bearings the diameter is increased to reduce deflection. This change in size is also used to serve as a locating shoulder.

As the blower is directly coupled, the only pressure upon the bearings is that due to the weight of the rotors. If the length of the bearing is taken as three times the diameter of the journal, a good proportion is obtained. The bearing shells are split horizontally and doweled to-gether. When the bearing case cap is taken off the bearings can be removed without disturbing the shaft. The bearing case is made oil tight by means of stuffing boxes. The detail of the bearing case is clearly shown in the general assembly drawing of the blower on page 103.

This drawing also illustrated the expanding inlet which is employed to avoid entrance losses and to insure rated capacity.

The arms which support the bearing case are made with a flat rectangular section so that they will offer the minimum amount of resistance to the passage of the air.

The volute opening is governed by the velocity of efflux. With the pre-determined pipe velocity of 70 feet per second, a discharge area of approximately 7 square feet is required. On account of employing high velocities in the impeller sufficient width is not available, when the casing is not wider than the wheel to secure the required outlet area and at the same time keep the overall dimensions of the volute in good proportion. The impeller in this case is twelve inches wide. Making the volute rectangular would require an opening 7 feet long for the proper discharge area. This would make the casing too large and too costly. To obtain the necessary width the sides are expanded at an angle of 45 degrees.

In turbo-blower practice, it is customary to choose a larger discharge velocity , and in that way keep down the dimensions of the casing. This, however, necessitates the use of an expanding section between the blower and the air duct, so that the velocity may be reduced and converted into pressure. This method makes the efficiency of the blower dependant upon the duct connections. Where space is limited, these may take such form that the change in velocity head will be wasted in eddies instead of increasing the pressure and in that way impairing the efficiency of the blower.

The beak of the volute is placed as close to the runner as possible, and in such a position that it divides the stream of air with the least disturbance and eddy losses. The contour of the volute is an equilateral spiral which gives a constant

increase of cross-sectional area.

The sides of a blower casing must have sufficient stiffness to withstand the internal air pressure. A pressure of one inch of water is equal to 5.2 pounds per square foot. To find the stress in the casing, consider a strip one inch wide at the discharge opening as shown in figure 23.

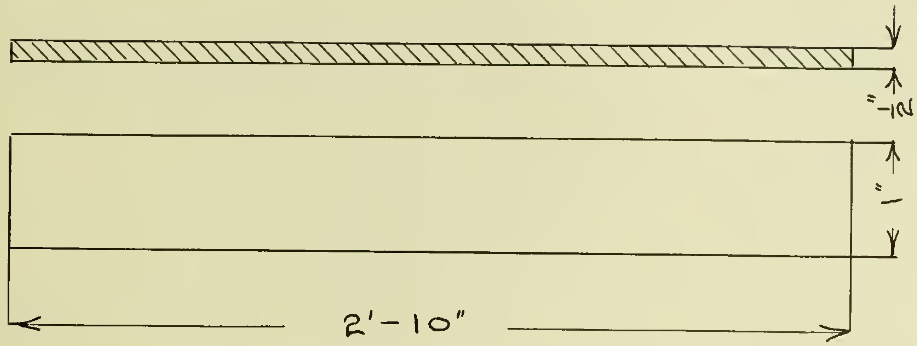


Figure 23.

Total pressure uniformly distributed with six inches water gage is

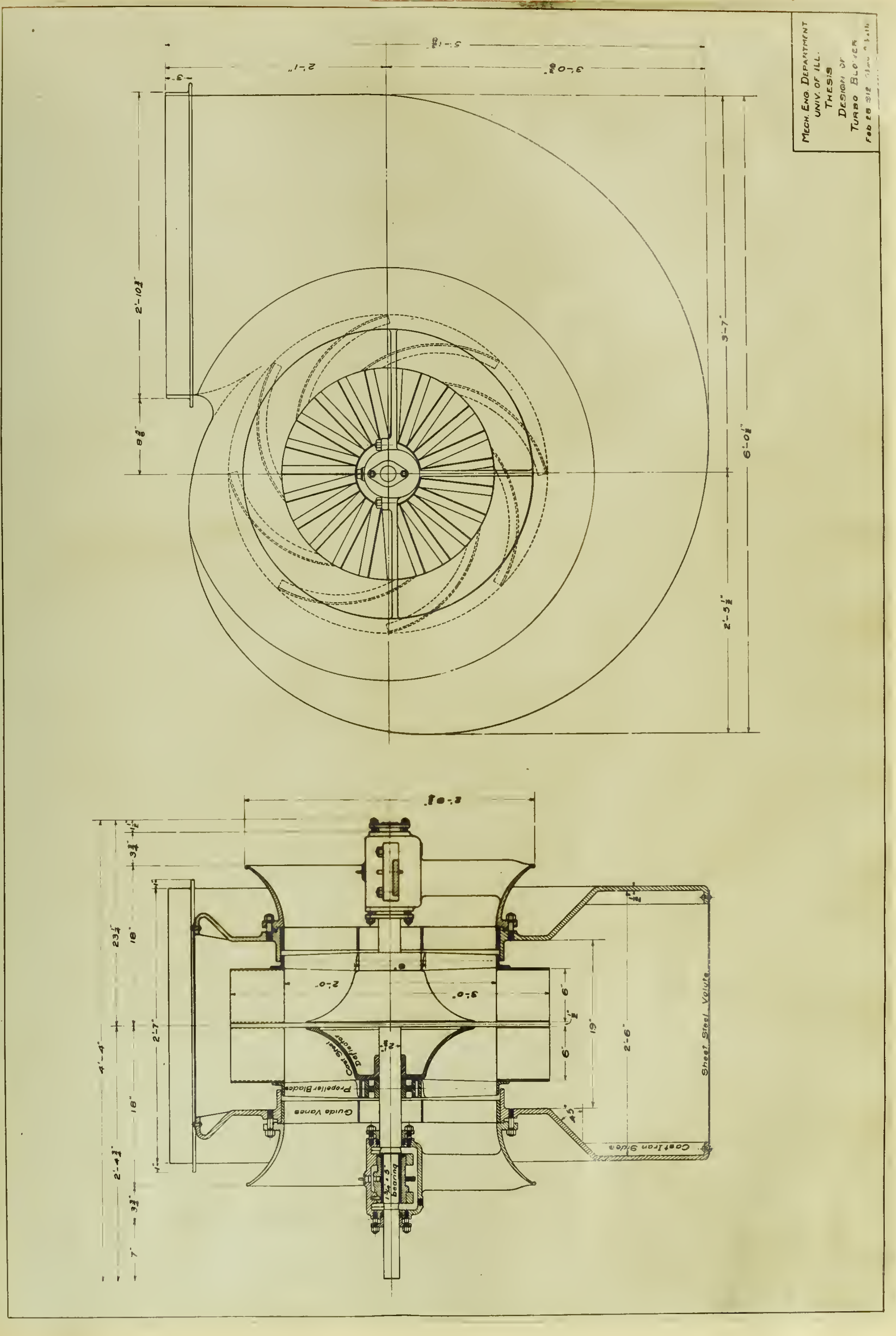
$$6 \times 5.2 \times \frac{2.835}{12} = 7.36 \text{ lbs.}$$

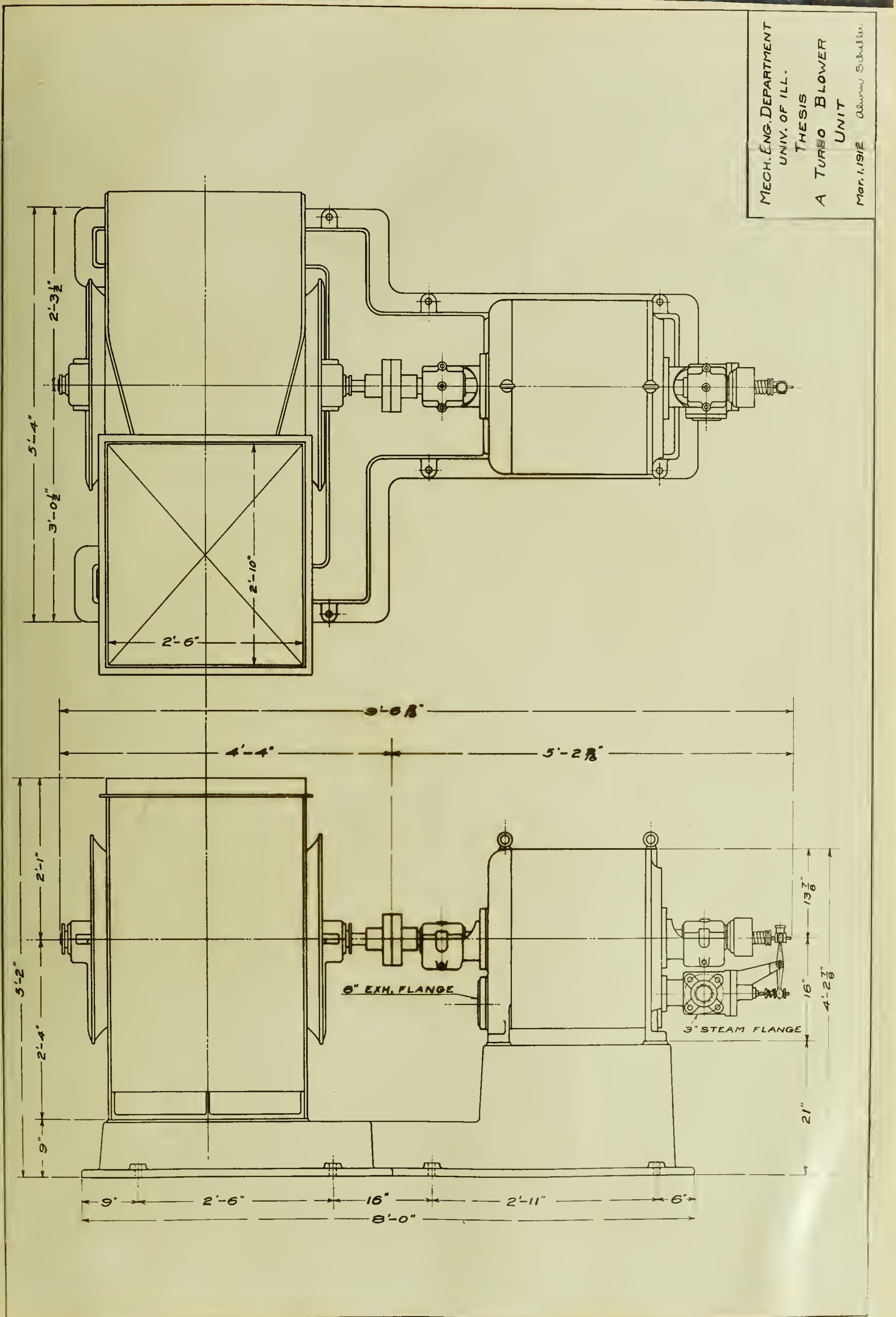
$$\text{Bending moment} = \frac{7.36 \times 34}{8} = 31.3 \text{ inch pounds}$$

$$\text{Section modulus} = \frac{1}{24}$$

$$\text{Fiber stress} = 31.3 \times 24 = 750 \text{ pounds per sq. inch}$$

In many blowers one dause for low efficiency is the "back blast" or leakage of air from the casing back into the suction. As this air has passed through the wheel, work has been done upon it and consequently the loss is directly proportional to the amount of leakage. To reduce this flow of air from the volute back into the impellor, the casing is so constructed that the clearance is made as small as possible and in addition the short circuited


Mech. Eng. Department
Univ. of Ill.
Thesis
Design of
Turbo Blower
Propeller Blades
Guide Vanes
Cast Steel Deflector
Sheet Steel Volute
Cast Iron Side
bearing
2'-10½"
3'-7"
6'-0½"
2'-5½"
2'-1"
3'-0"
2'-0"
3'-0"
2'-6"
2'-7"
4'-4"
2'-4½"
23½"
18"
19"
6"
7"
45°


MECH. ENG. DEPARTMENT
UNIV. OF ILL.
THESIS
A TURBO BLOWER UNIT
Mar. 1, 1912
6" EXH. FLANGE
3" STEAM FLANGE
8'-0"
4'-4"

air must flow around a sharp corner which exerts a marked throttling action.

To insure good operation, the blower and turbine should be mounted on a rigid bed plate. The drawing on page 104 shows an assembly view of the turbine and blower. The machines are connected with a flexible coupling. When accurately assembled, and doweled into position so that all possibility of change of alignment is eliminated a vibrationless operation is obtained which lasts throughout the life of the machines.

The turbine shown in this drawing is a seven stage machine built by the Kerr Turbine Company. The calibration curve shown on page 13 is typical of this type of turbine and may be taken to represent the performance of the machine selected.

VIII - METHOD OF TESTING TURBO BLOWERS.

When a blower unit is to be installed to operate under the service conditions outlined in the beginning of this discussion its characteristic curves should be determined. By the aid of these curves, the operating engineer will be enabled to not only secure a sensitive regulation of the air supply, but also to insure the most efficient performance of the unit over its entire range of operation. The only method of determining these characteristics accurately is by means of a reliable test.

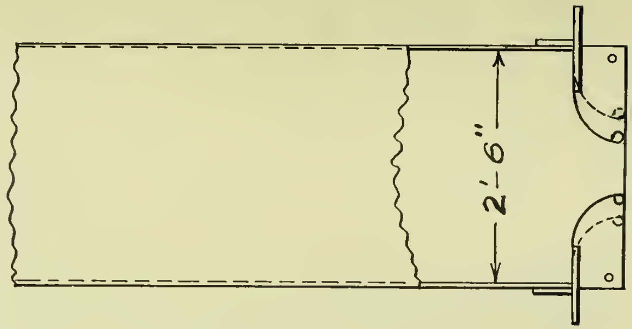
If a blower delivers air to a duct, when operating at constant speed, the discharge or quantity of air delivered will vary with the resistance. As the resistance is changed, the delivery and pressure will both vary. However, for each particular pressure, there will be a definite delivery. If the resistance at the end of the duct were varied from zero to shut off by successive changes and the pressure and quantity measured for each point, a complete index of the capacity and pressure variation at constant speed is secured. The values thus obtained may be plotted with delivery as abscissae and the pressure as ordinates. A smooth curve drawn through these points from no delivery to full delivery is known as the pressure characteristic at constant speed. When this curve has been determined, the discharge corresponding to any pressure can be found.

If at the same time, the horse power input is measured a horse power capacity curve can be drawn. From this data the mechanical efficiency can be calculated. A curve through these points, also plotted with capacity as abscissae, is known as

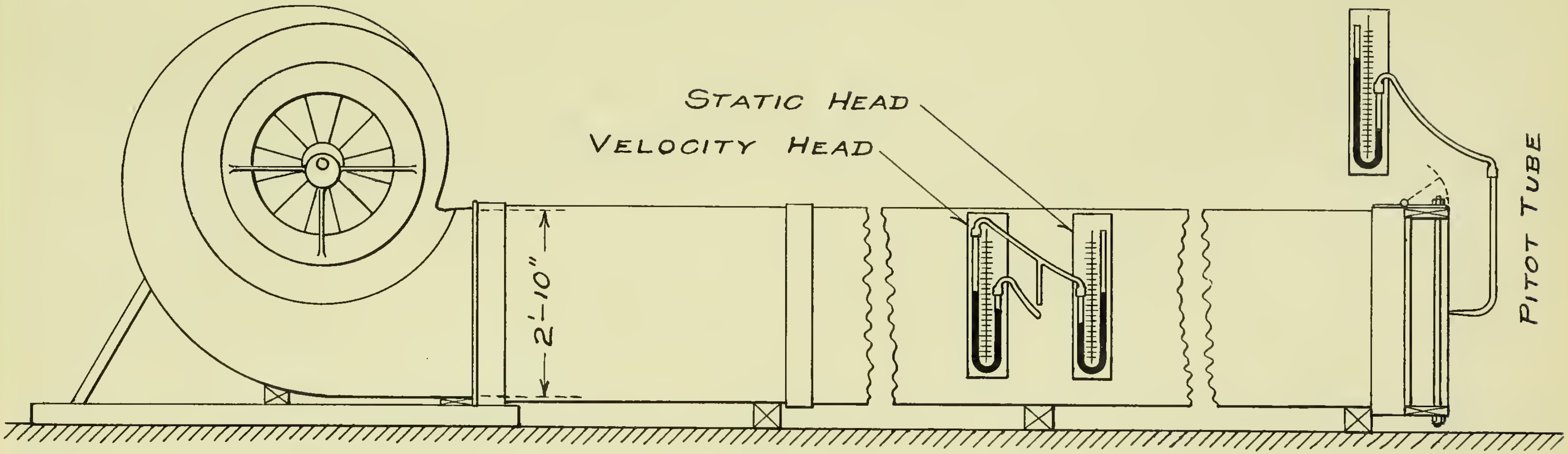
the efficiency characteristic. When these curves have been drawn, the brake horse power efficiency and pressure can be read off at once for any capacity. Furthermore, the economical range of the blower is established, and the variation in brake horse-power from shut-off to full delivery is determined. Illustrations of such characteristics are given in connection with the analysis of the experiments upon blowers. A blower test means nothing more or less than determining these characteristics for a series of speeds covering the practicable range of operation.

The data required is therefore, the power input , the pressure and the delivery. On page 108 adiagramatic drawing of a blower testing outfit is shown. The blower is driven directly by a steam turbine. The resistance is varied by changing the gates at the end of the duct. These are fixed in one position while simultaneous readings of brake horse power, pressure and delivery are taken. After changing the gates to a new position, the readings are repeated; the blower operating meanwhile at constant speed. In this way the entire range from full delivery to shut off is obtained.

In the discussion of the small turbine, it was shown that under proper conditions of operation, it could be employed for measuring the power input to the driven machine. If a turbine has been calibrated and the conditions of back pressure are identical with those obtaining during the calibration, the steam pressure on the first stage nozzles will at once give the brake horse power. It must be emphasized, however, that initial steam and exhaust conditions cannot be varied from the original, if reliable results are expected. If dry steam was used for the turbine test, the same steam must be used for the blower test. With these sim-



TOP VIEW OF RECTANGULAR NOZZLE



ELEVATION OF BLOWER TESTING APPARATUS

ple precautions a ready and convenient method is at hand for determining the brake horse power. For each reading, it is only necessary to note the steam pressure on the first stage with a calibrated gage.

The compression can also be easily and accurately measured. It is becoming common practice to give the static pressure in inches of water for the reason that this is the most convenient method of measurement. A common glass U tube is connected to an opening in the duct as shown in the drawing on page108. The difference in level in the two branches of the tube registers the pressure in the duct. If the pressure is small, greater accuracy can be obtained by inclining the U tube and also by using a liquid of lower specific gravity than water, as for example, gasoline.

Measuring the quantity of air delivered, however, presents more serious difficulties. Many tests have been reported, especially upon mine ventilators, in which the delivery was determined with an anemometer. As the results obtained with this instrument depend entirely upon its calibration, it cannot be applied with the expectation of securing reliable values of the discharge.

A more accurate method of measuring air is by means of the Pitot Tube. This instrument is used essentially for measuring the velocity of the air. Having found the velocity, the quantity flowing is easily determined. The Pitot Tube may be used in two ways; either to measure the velocity in the duct, or to measure the velocity of the air issuing through a nozzle at the end. As these methods for measuring air have such an important bearing upon blower trials, they will be discussed in greater detail.

If a tube is inserted in a stream of air so that the stream lines impinge normally upon the place of its opening, and the other

end be connected to a suitable gage, the pressure recorded will be that due to the static head in the duct plus that due to the velocity head. With a tube inserted at right angles to the stream lines the pressure recorded will be that due to the static head alone. The difference therefore of these readings will be the velocity head. If both tubes are connected to the opposite branches of a pressure gage, the reading will be the difference between total and static head or the velocity head. To go a step farther, both tubes can be combined in one, as shown in figure 25.

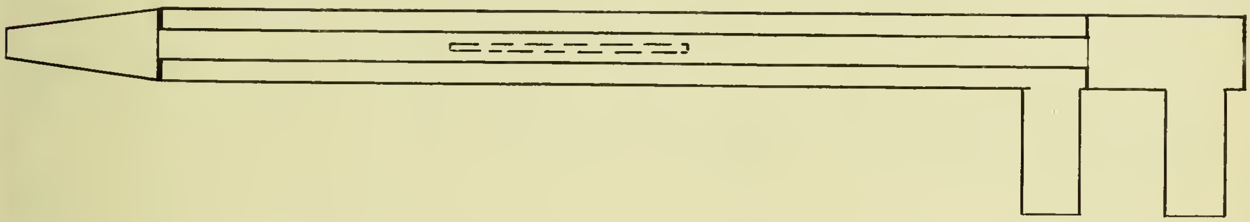


Figure 25.

When this tube is immersed in a stream of air, the air immediately in front of its impact opening is gradually brought to rest increasing its pressure and temperature. The pressure in the stream of air is measured through the side slots of the tube. If then both ends of this tube are connected to a differential gage, the velocity head is at once determined. The following thermodynamic theory of the Pitot Tube is due to D.W. Taylor U.S.N.

The fundamental equation connecting pressure, velocity, density and temperature of a perfect gas in steady motion is

$$\frac{V_1^2 - V_2^2}{2g} = \frac{k}{k-1}\,\frac{p_1}{D}\left[1 - \left[\frac{p_1}{p_2}\right]^{\frac{k-1}{k}}\right]$$

Here v_1 = velocity in feet per second where pressure is p_1 lbs. per square foot.

p_2 = pressure in lbs. per square foot at any other point

v_2 = velocity, when pressure is p_2

D = weight per cubic foot, when pressure is p_2

k = $\frac{CP}{CV}$ = 1.408

g = acceleration due to gravity

Let p_2 = pressure at impact opening of Pitot Tube. Then v_2 = 0. Also take p_1 as the pressure of the unchecked stream of air measured by the manometer connected to the pressure openings on the side of the Pitot Tube.

Let pa Da Ta be the pressure, weight, and absolute temperature of the atmosphere;

T_1 and T_2 absolute temperatures corresponding to p_1 and p_2.

Then $\frac{pa}{Da\ Ta} = \frac{p_2}{D_2 T_2} = \frac{p_1}{D_1 T_1} = x$

The compression from p_1 to p_2 is adiabatic. Therefore

$$\frac{T_2}{T_1} = \left[\frac{p_2}{p_1}\right]^{\frac{k-1}{k}}$$

Therefore

$$\frac{p_2}{P_2} = x\,T_2 = x\,T_1\left[\frac{p_2}{p_1}\right]^{\frac{k-1}{k}}$$

$$\frac{V_1^2}{2g} = \frac{k}{k-1}\,x\,T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right]$$

Let

$$p_2 = p_1[1+y]$$

Then

$$\frac{V_1^2}{2g} = \frac{k}{k-1}\,x\,T_1\left[(1+y)^{\frac{k-1}{k}}\right]$$

Expanding and reducing

$$\frac{V_1^2}{2g} = xT_1\left[y - \frac{y^2}{2k} + y^3\frac{1+k}{6k^2} - y^4\frac{(1+k)(1+2k)}{24\,k^3}\right]$$

Now $k = 1.408$

$$\frac{1}{2y} = .355$$

Write a first approximate formula

$$\frac{v_1^2}{2g} = xT_1\left[y - \frac{y^2}{2k}\right] = xT_1\left[\frac{p_2 - p_1}{p_1} - .355\left[\frac{p_2 - p_1}{p_1}\right]^2\right]$$

A second approximate formula

$$\frac{v_1^2}{2g} = xT_1 y = xT_1\left[\frac{p_2 - p_1}{p_1}\right]$$

Taking $g = 32.2$

$x = 53.25$

$T_1 =$ 531 degrees absolute, or 70 degrees farenheit

We have, by computation

y	$\frac{p_2}{p_1}$	Velocity v_1 Exact	First Approx.	Second Approx.
.0014	1.0014	50.45	50.45	50.46
.0055	1.0055	99.91	99.92	100.01
.01	1.01	134.61	134.62	134.86

The velocity of air in ducts is rarely taken over 70 feet per second and up to that velocity, the simple equation

$$\frac{v_1^2}{2g} = xT_1\left[\frac{p_2 - p_1}{p_1}\right]$$

is practically exact. Even at a velocity of 100 feet per second the error of this formula as compared with the exact equation is but one tenth of one percent

Now $x = \frac{p_1}{D_1 T_1}$

Substituting in the simple equation above

$$\frac{v_1^2}{2g} = \frac{1}{D_1}(p_2 - p_1)$$

$$v_1^2 = \frac{2g}{D_1}(p_2 - p_1)$$

D_1 is the weight per cubic foot of the air in the pipe. Its

value depends upon the pressure, temperature and humidity of the air in the duct. If these factors have been determined, the weight per cubic foot of air can be calculated from the equation given in the Smithsonian Tables.

$$D = \frac{.080723}{1 + .0020389\,(t + 32)} \times \frac{b - .378e}{29.921}$$

Here t = temperature degrees farenheit

b = height of barometer

e = vapor pressure in inches of mercury

The values for D should be computed and tabulated so that they can be referred to when computing a blower trial. For finding the velocity in the pipe, the simplest equation is

$$v_1 = \sqrt{\frac{2g}{D_1}}\,(p_2 - p_1)^{\frac{1}{2}}$$

If a differential gage is used which measures at once the difference of pressure, then

$$v_1 = \sqrt{\frac{2g}{D_1}\,p}$$

Where $p = p_2 - p_1$. Where the velocity has been determined, the quantity follows by multiplying by the cross sectional area. A Pitot Tube directed against the direction of flow in the duct will register both the static and the velocity head. In other words, it is said to represent total head. As the air issues from the end of the duct, the static pressure has all been absorbed in creating velocity. A Pitot Tube therefore, inserted in the stream issuing through a nozzle at the end will only register velocity head as there can be no lateral pressure in the stream at this point. However, the reading obtained here will be equal to the one obtained in the duct, because the velocity head at the end of the nozzle is equal to the velocity head plus the static head in the duct. To determine the velocity of efflux from the nozzle, it is only necessary to take one reading as

compared with two in the duct.

A seeming element of doubt enters when the discharge is to be computed. Different forms of nozzles will give different contractions of the jet. The accuracy of the calculated delivery therefore depends upon the nozzle coefficient assumed. For testing large blowers the nozzle shown on page 108 is a convenient one to use. The gates can be moved so as to vary the area. The sides are formed of sheet iron rounded so as to give a full stream. The sheet iron being rolled so that the air has tangential entrance and exit. These sides are clamped into position with hinged doors above and below. It would be natural to suppose that the air would issue with considerable contraction from such a nozzle. However, this form of end nozzle has been employed in a large blower trial where a double Pitot was used in the duct. When the results were computed, the discharge calculated from the readings taken with the double tube was in every instance larger than that found by the readings taken at the nozzle and assuming a full jet with no contraction. Furthermore, when plotted for obtaining the head capacity curve, the points calculated with the latter method defined a smoother curve.

As a matter of convenience and cost, the advantage lies with the end nozzle method. To obtain accurate results by measuring the velocity in the duct it must be of sufficient length to insure parallel stream lines and no eddies. Good practice prescribes a length equal to twenty times the diameter of the discharge opening. For large blowers, this length may reach ninety feet. With the end nozzle such a long duct is not necessary. The friction losses are also minimized, which of course raises the efficiency of the blower.

To illustrate the application of the method of testing blowers described here, a set of readings will be assumed for one condition and the necessary computations made. Consider the blower previously designed to be fitted for testing, as indicated by the outline drawing on page 108; the air being measured both in the duct and at the end nozzle. For reliable results, the weight of a cubic foot of air under test conditions must be known. Consequently temperature, barometer and hygrometer readings must be taken. The humidity and thus the vapor pressure are found. The weight of air can then be computed from the Smithsonian equation. These computations will not be repeated and we shall take the weight of air as .0755 lbs. per cubic foot.

By measurement area of duct = 7.08 square feet

By measurement area of end nozzle = 2.8 square feet

Readings obtained

Static gage in duct 6 inches of water

Velocity " " " 1.1 " " "

End Pitot 7.1 " " "

Steam pressure on first stage nozzles 100 lbs. per square in.

Speed 1750 R.P.M.

Now, a pressure of one inch of water is equal to 5.2 lbs. per sq.ft.

Velocity in duct $= \sqrt{\dfrac{2g \times p}{D}}$

Substituting

$$v = \sqrt{\frac{2 \times 32.2 \times 1.1 \times 5.2}{.0755}} = 70 \text{ ft. per sec.}$$

Discharge = 70 x 7.08 x 60 = 29,800 cubic ft. per minute

Velocity through nozzle is

$$v_1 = \sqrt{\frac{2 \times 32.2 \times 7.1 \times 5.2}{.0755}} = 177 \text{ ft. per sec.}$$

Discharge = 177 x 2.8 x 60 = 30,000 cubic feet per minute.

Total pressure by both gages is 7.1 inches of water.

Air horse power $= \frac{30,000 \times 7.1 \times 68.3 \times .0755}{33,000} = 34.3$

From the calibration curve of the turbine 100 pounds pressure at 1750 r.p.m. gives 53 brake horse power.

Mechanical efficiency of blower $\frac{34.3}{53} = 63\%$

IX -- OPERATING CHARACTERISTICS OF PROPOSED BLOWER.

A centrifugal pump or blower may be operated over a wide series of speeds without altering the maximum efficiency very materially. However, for each speed there will be a particular capacity at which the maximum occurs. Naturally the greater the speed the larger the discharge for the most efficient performance. The head obtained at the best discharge also varies as the speed changes. There is a very definite and fixed relation between the economical head and capacity at one rotative speed, and the corresponding economical head and capacity at another speed. When expressed by means of an equation, this relation takes the form $Q^2 = kh$. Here Q represents the capacity and h the head, while k is some constant. This is the equation of a parabola and may be used to find the economical head corresponding to any given discharge or vice versa. The constant k must be determined however, before the equation can be applied. There is only one method of finding the value of k; the head and discharge corresponding to the point of maximum efficiency for one speed must be known. By substituting the known values of Q and h the value of k is determined. If this equation is plotted graphically with capacity as abscissae and pressure as ordinates, the resulting curve will intersect the pressure characteristics for a series of speeds at points of maximum efficiency.

The equation $Q^2 = kh$ may be derived in the following way. Repeated tests both upon centrifugal pumps and blowers have shown that the discharge angle for maximum efficiency is independent of the rotative speed. Or stated in another way, the discharge angle for best efficiency at any speed is a constant.

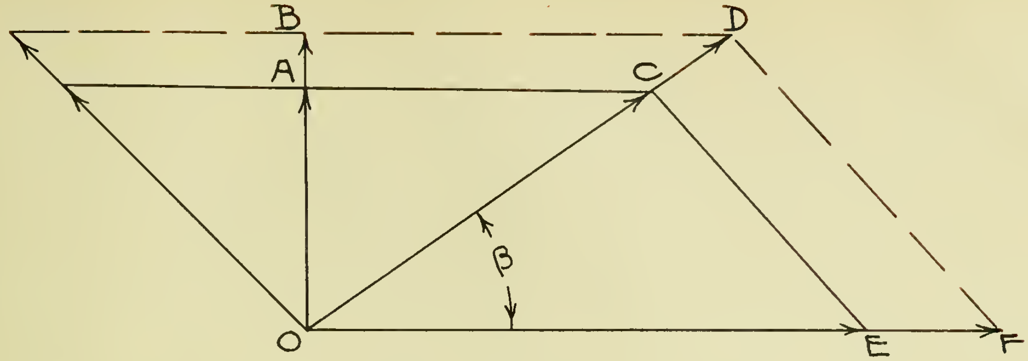


Figure 26

Figure 26 shows the exit velocity diagram of a blower at maximum efficiency for the peripheral speed OE. Let the speed increase to OF; as previously stated the angle β must remain constant. If, therefore, OA represented the economical capacity for the speed OE, then OB represents the economical capacity for speed OF.

Let OA $= Q^1$ given economical capacity

OC $= v^1$ velocity of discharge

OB $= Q$ economical capacity for any speed

OD $= v$ corresponding discharge velocity

Then by similar triangles

$$OB = OA \times \frac{OD}{OC}$$

$$Q = Q^1 \times \frac{v}{v^1}$$

But the velocities vary as the square root of the heads. Therefore

$$Q = Q^1 \times \frac{\sqrt{h}}{\sqrt{h^1}}$$

$$Q^2 = \frac{\overline{Q^1}^2}{h^1} h$$

Let $\frac{\overline{Q'}^2}{h'} = k$ a constant. Then

$$Q^2 = kh$$

The equation just obtained while valuable in showing the effect of changes of speed upon the head and capacity gives no aid in predetermining the operating characteristics. The only reliable

method of predicting performance is by a comparison with a similar machine that has been built and tested. As a general rule, it will be impossible to find a machine of the same capacity, head and speed so that a comparison can be made directly. If the impellers have similar vanes, they may be of different size, designed for different capacities; if the heads correspond, the speeds may vary and so on. A method has been devised by means of which two impellers may be compared regardless of diameter, vane angles, capacity or speed. This is known as a comparison on the basis of "Unit Speed."

The unit speed N_u is the speed, under one inch head, of a wheel of the same design as the one under consideration and of such size as to develop one air horse power. At or near maximum efficiency, the following laws govern the action of centrifugal blowers.

1 - The head varies as the square of the speed

2 - The capacity varies directly as the speed.

3 - The power varies as the head raised to the 3/2 power

4 - The power varies directly as the square of the diameter of the impeller in homologous runners.

Let

N = speed of any impeller

P = power of same

D = diameter of same

h = head on same

n_u = unit of head

P_u = " " power

N' = speed of runner for unit head

P' = power " " " " "

Du = diameter of homologous runner developing unit power under unit head.

Applying the foregoing laws; then

$N' : N = h_u : h, \quad N' = N \frac{n_u}{n}$ ---------- a

$P' : P = h_u^{3/2} : h^{3/2}, \quad P' = P\left(\frac{n_u}{h}\right)^{3/2}$ ---------- b

$P' : P_u = D^2 : D_u^2, \quad D_u = D \frac{P_u}{P'}$ ---------- c

$\frac{N'}{N_u} = \frac{D_u}{D}$ ---------- d

Substituting equations a, b and c in equation d

$$\frac{N\sqrt{\frac{h_u}{h}}}{N_u} = \frac{D\sqrt{\frac{P_u}{P\left(\frac{h_u}{h}\right)^{3/2}}}}{D}$$

Simplifying

$$N_u = \frac{N \times \sqrt{P} \times h_u^{5/4}}{\sqrt{P_u} \times h^{5/4}}$$

With Pu as one horse power, and h_u as one inch, this becomes

$$N_u = \frac{N\sqrt{P}}{h^{5/4}}$$

With the aid of this equation, the unit speed of any impeller may be determined when the speed, head and delivery are known.

Since the unit speed is a measure of completer performance, both as to speed and capacity, it supplies a good basis of comparison. Attention is called to the fact that vane angles, diameter, and peripheral speed and other details of design are excluded from this equation. As capacity enters as a factor, large and small blowers may be compared directly.

Wheel No. 2, tested by Mr. Guy, delivered 5250 cubic feet of air per minute against a static pressure of five inches of water with a speed of 3600 r.p.m.

Here P = 4.15 air horse power

N = 3600

h = 5

The unit speed of this runner at maximum efficiency was therefore

$$Nu = \frac{3600 \times \sqrt{4.15}}{5^{5/4}} = 976$$

The efficiency at this unit speed was 58%. In the same way, the unit speed for other points at the same speed may be computed. The values thus obtained have been plotted in the form of a curve shown on page 123. Here unit speed has been taken as abscissae and efficiency as ordinates. This curve does not intersect the axis of zero efficiency at any finite unit speed. At full delivery the head becomes zero, and the unit speed becomes infinite.

These results will be used to predict the performance of the proposed design, As Mr. Guy's values of the mechanical efficiency are based upon static head, a comparison must be made upon that basis. For the proposed design, the static pressure is equal to six inches of water at rated capacity.

P = 28.5 air horse-power (based on static pressure)

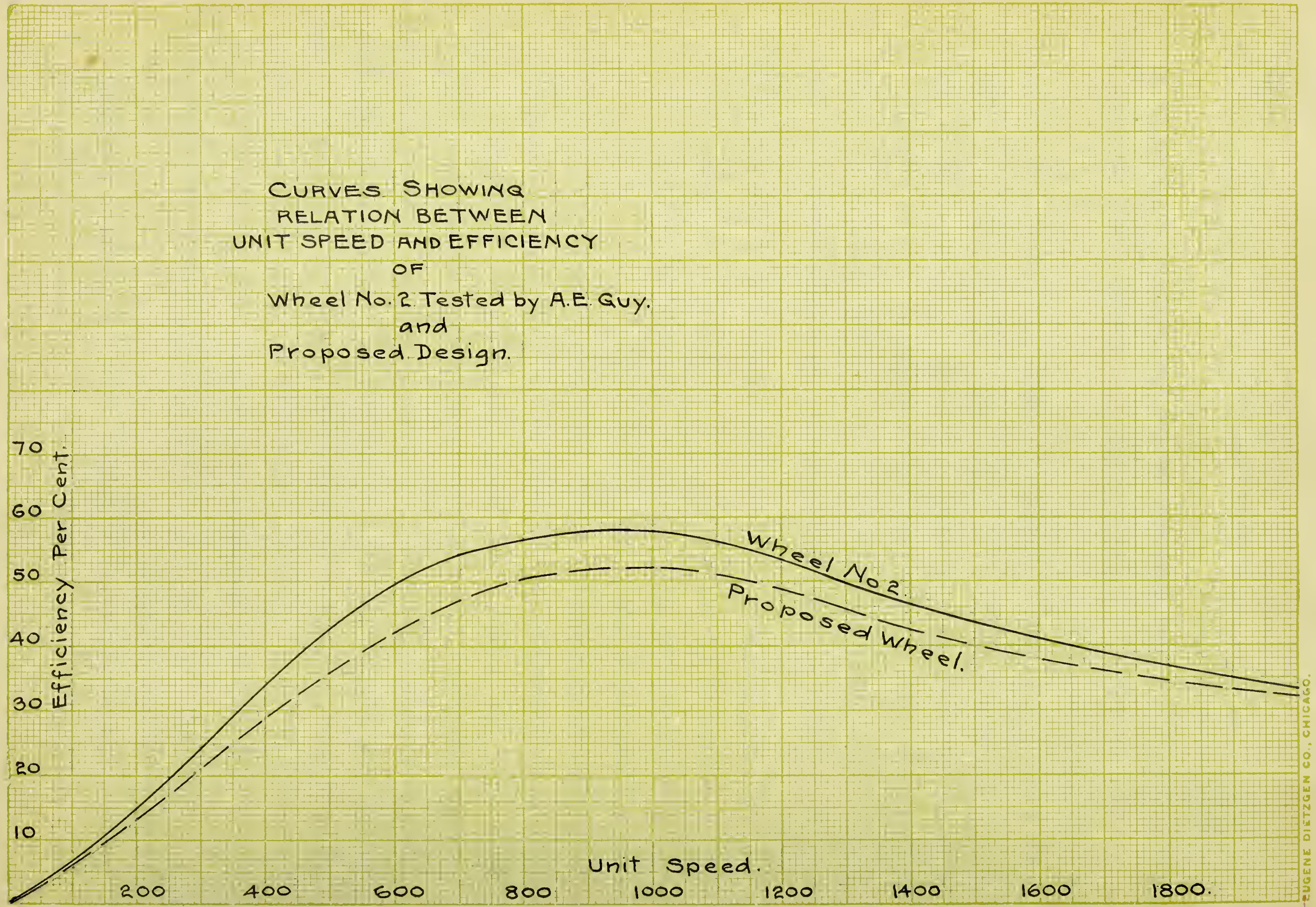
N = 1750 r.p.m. n = 6 inches of water

$$Nu = \frac{1750 \times \sqrt{28.5}}{6^{5/4}} = 1000$$

The proposed design has therefore a unit speed which is but very little higher than that of Wheel No. 2 of Mr. Guy's experiments. They can be considered as homologous runners. The maximum efficiency of the proposed design based upon static pressure has been chosen as 52%. This value was checked by applying Innes equation for mechanical efficiency. With the additional entrance losses that are incurred, on account of the mechanical means adopted for bringing the air into the wheel, the efficiency cannot be as high as that of Wheel No. 2. However, as these are homologous runners, they must have similar efficiency curves. As the proposed wheel is a slightly higher speed runner than Wheel No. 2, its efficiency curve on a unit speed base will drop a trifle less slowly as the unit speed increases. The efficiency curve for the proposed wheel can

now be drawn, as shown on page 123. The maximum point being established at a unit speed of 1000 and an efficiency equal to 52%. It must have a form similar to that of Wheel No. 2 with the exception noted above.

The efficiency curve thus drawn is of little value until the head characteristic has been established. Thus far only one point upon this curve is known, namely the point of maximum efficiency. The general form of this curve is at once fixed by the shape of the vanes. Referring to the velocity polygon at exit, it is evident that as the capacity decreases, the absolute velocity increases, approaching the rim velocity as a limit. This increase in discharge velocity causes a rising head toward shut-off, where the maximum pressure will be obtained. The static no delivery head usually bears some relation to the peripheral velocity, which is determined by the form of runner. The shut-off pressure of Wheel No. 2 of Mr. Guy's experiments was 11.5 inches of water. The peripheral velocity of the impeller was 283 feet per second. The theoretical head due to this velocity is 18.4 inches of water. With this impeller, therefore, the shut-off pressure was 63% of the theoretical. Full delivery or capacity at zero head may be expressed as a function of the capacity at maximum efficiency. There appears to be no exact mathematical relation between these two quantities. However, when maximum efficiency has been reached, the pressure begins to fall off rapidly and full delivery is soon obtained. The radial velocity of the air from the wheel has an important bearing upon this ratio. If this velocity has been taken small, there will be a greater difference between the maximum and economical deliveries than if it was assumed large. In a well designed impeller, the capacity at full delivery will vary between


CURVES SHOWING
RELATION BETWEEN
UNIT SPEED AND EFFICIENCY
OF
Wheel No. 2 Tested by A.E. Guy.
and
Proposed Design.
Efficiency Per Cent.
70
60
50
40
30
20
10
Unit Speed.
200
400
600
800
1000
1200
1400
1600
1800.
Wheel No 2.
Proposed Wheel.
EUGENE DIETZGEN CO., CHICAGO.

1.35 and 1.5 times the capacity at greatest efficiency. If this ratio is found to be as high as 2, as the results of some published tests indicate, it signifies that the rotor is too large for the required delivery.

With this data, we are enabled to draw a close approximation to the head capacity characteristic at the constant speed of 1750 r.p.m. This curve must give the static pressure only as our efficiencies have been taken on that basis. This curve therefore passes through the desired pressure of six inches of water at 30,000 cubic feet per minute. The shut-off pressure for this impeller will bear a higher ratio to the peripheral velocity than exhibited by Mr. Guy's wheel No. 2 on account of the action of the propeller blades which will themselves put up head as the delivery is throttled. The static no-delivery-head will therefore be taken as .75 of the head theoretically due to the rim velocity.

Rim velocity at 1750 r.p.m. = 275 feet per second

Head due to rim velocity = 17.4 inches of water

Shut-off pressure = .75 x 17.4 = 13 inches of water

On account of the large capacity for handling air of a propeller wheel, the full delivery of this blower will be close to the maximum ratio of well designed impellers. If the maximum capacity is taken as 1.45 times the economical the estimate will be conservative. Wheel No.2 of Mr. Guy's experiments gives the same ratio and it had no mechanical device for assisting the flow of air into the wheel

Full delivery = 1.45 x 30,000 = 43,500 cubic feet per min. The three points thus obtained define the head capacity curve which is shown on page 126.

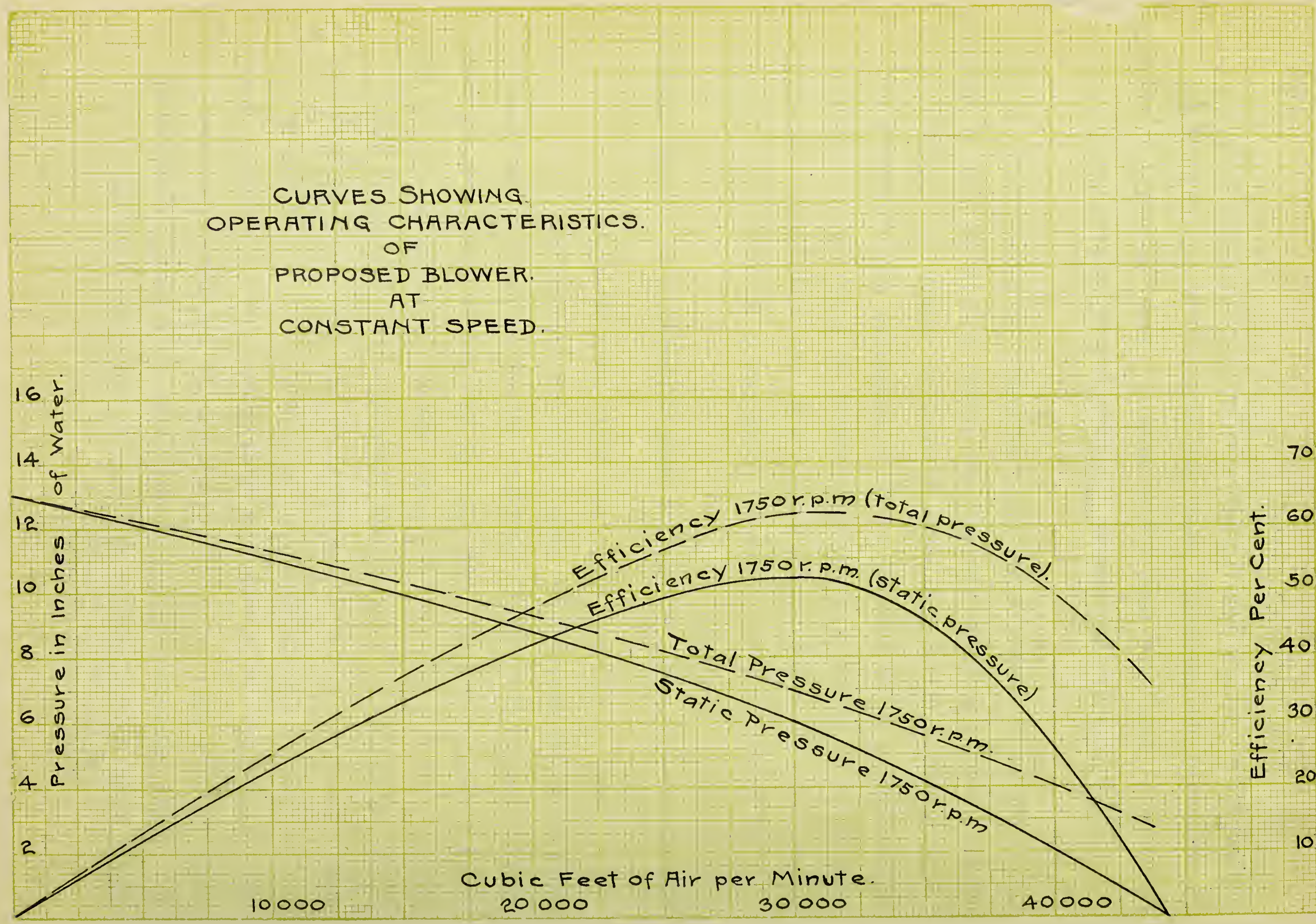
With the aid of the unit speed efficiency curve already drawn, the efficiency characteristic can now be determined. The efficiency at rated conditions is known to be 52%. To find the efficiency at any other point, we proceed as follows. At 25,000 cubic feet per minute, the pressure is 7.5 inches of water. This gives an air horse power of 29.6. Substituting in the equation for unit speed

$$n_u = \frac{1750 \times \sqrt{29.6}}{7.5^{5/4}} = 743$$

By reference to the unit speed efficiency curve, the efficiency corresponding to a unit speed of 743 is found to be 49%. In the same way the efficiencies for other capacities were computed and the values thus obtained plotted graphically on the same sheet with the head characteristic. A smooth curve through these points gives the efficiency characteristic.

The results thus obtained for static pressure are now transformed to total pressure. At rated discharge the velocity at the volute opening of the blower is 70 feet per second which corresponds to a pressure of 1.1 inches of water. The total pressure at this delivery is therefore 7.1 inches of water. In a similar way the velocities and corresponding pressures were found for other deliveries and the total pressure characteristic drawn. As shown on page 126 the two curves have the same shut-off value as the velocity head is zero at this point. Naturally as the delivery increases the deviation between the total and static head characteristics becomes larger.

The efficiency curve based upon total pressure is computed from that based upon static pressure by simple proportion. For example, at rating, static pressure is 6 inches, total pressure 7.1 inches, and efficiency 52%. The desired efficiency is there-



CURVES SHOWING
OPERATING CHARACTERISTICS.
OF
PROPOSED BLOWER.
AT
CONSTANT SPEED.
Pressure in Inches of Water.
Efficiency Per Cent.
Cubic Feet of Air per Minute.
Efficiency 1750 r.p.m (total pressure).
Efficiency 1750 r.p.m. (static pressure)
Total Pressure 1750 r.p.m.
Static Pressure 1750 r.p.m
2
4
6
8
10
12
14
16
10
20
30
40
50
60
70
10000
20 000
30 000
40000

fore

$$\frac{7.1}{6} \times .52 = 62\%$$

In this way the values for the entire curve are obtained.

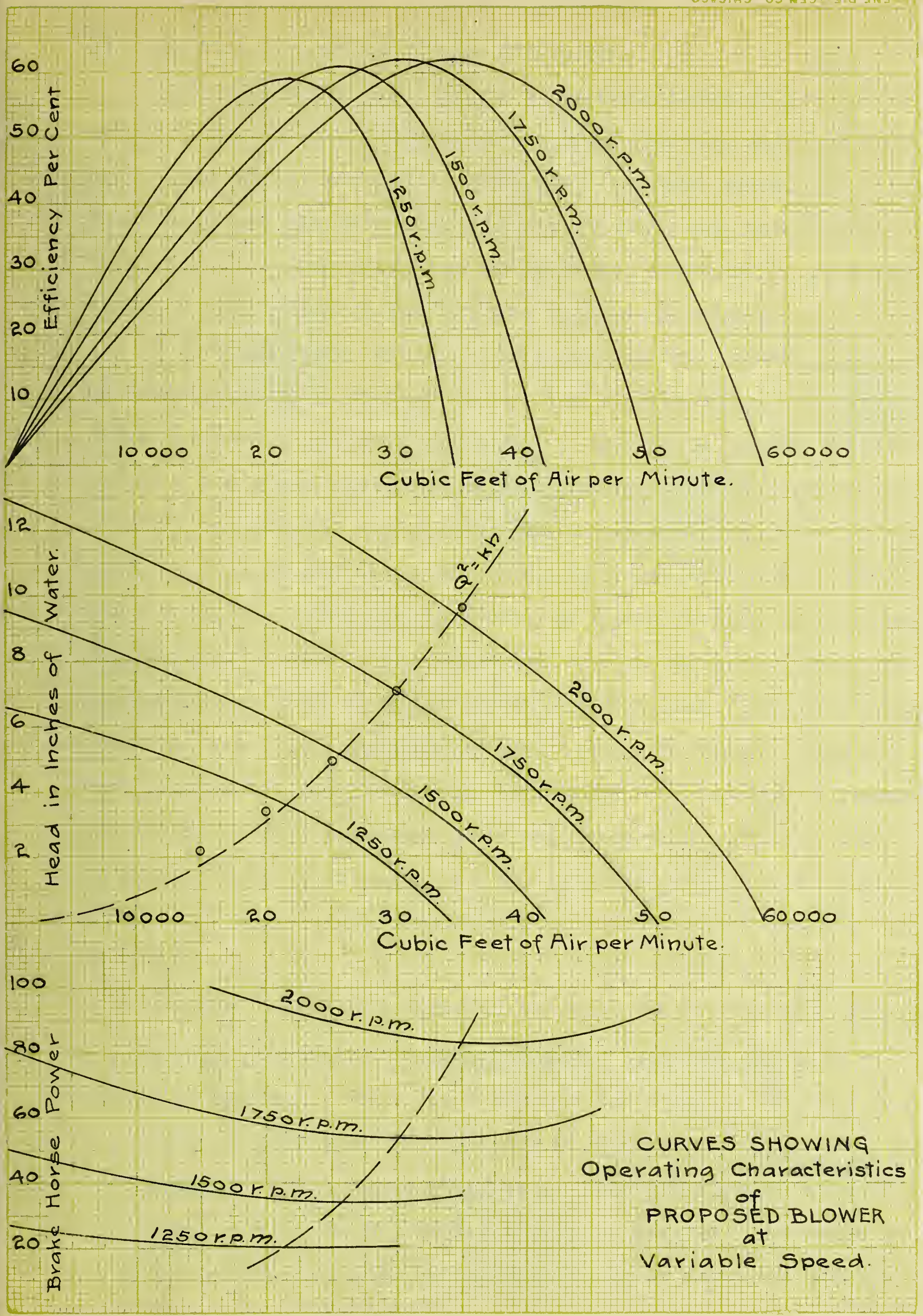
Those characteristics are replotted to a smaller horizontal scale on page 128. It is now necessary to determine the same characteristics for a series of speeds which will entirely cover the complete range of operation of this blower. The heads and capacities for highest efficiencies at all speeds are determined by the equation $Q^2 = kh$. One point on this parabola is known namely $Q = 30,000$ and $h = 7.1$. These values determine the constant k, and the parabola may then be plotted as shown. At or near the maximum efficiency the capacity varies directly as the speed. The economical deliveryfor 1500 r.p.m. will therefore be

$$\frac{1500}{1750} \times 30,000 = 25,700 \text{ cubic feet per minute}$$

From the parabola $Q^2 = kh$, the head corresponding to this capacity is 5.2 inches of water.

By using the same method employed for rated speed, the characteristics for the speed of 1500 r.p.m. are determined. After the head curve is drawn it is only necessary to find the unit speeds for different capacities in order to establish the efficiencies. The unit speed efficiency curve assumed is independant of the rotative speed. For this reason, therefore, this curve can be applied for finding the entire series of efficiency characteristics of this blower.

Proceeding in this manner the head and efficiency curves for 1250, 1500, 1750, and 2000 r.p.m. were all drawn as shown on page128 By combining the capacity, head, and efficiency, the brake horse power curves for the same rotative speeds can be determined and



CURVES SHOWING
Operating Characteristics
of
PROPOSED BLOWER
at
Variable Speed.

are shown on the bottom of the same page.

This series of curves forms a complete index of the performance of this blower. The rotative speed for any delivery and head is found by interpolating between the head characteristics. When the speed has been found, the efficiency and brake horse power for the given condition of operation can readily be determined by interpolation between the respective curves.

The performance of the blower has now been established. The important question, whether or not the machine meets the requirements originally proposed now arises. Referring to the curve on page 9 the total pressures corresponding to various rates of delivery and stoking are given. The necessary pressures for capacities from 10,000 to 35,000 cubic feet per minute as determined from this curve are indicated by small circles on the head, capacity curve sheet. The speed for any air supply is now fixed. For example at 150% of boiler rating 25,000 cubic feet of air are required at a total pressure of five inches of water. From the head characteristics the blower is required to operate at approximately 1500 r.p.m. to supply the proper amount of air for this particular rate of stoking. In this way the speed of the turbo blower unit can be varied as required by the demand upon the boilers for steam.

The most striking feature of the requirements is that <u>the points of necessary operation lie almost exactly upon the parabola of maximum efficiency</u>. This means that as the speed of the turbine is changed to meet the varying rates of firing <u>the blower operates continously at its maximum efficiency</u>. The superiority of such a unit over one in which the air supply is regulated by means of a blast gate needs no further argument. It is for such service

where varying speed is almost essential that the small steam turbine is particularly well adapted.

The maximum overall efficiency of the unit is obtained when the blower is operating at 2000 r.p.m. with an efficiency of 62%. The shaft horse-power delivered by the turbine at this point is 86. To develop this horse-power a steam pressure of 140 lbs is required upon the first stage nozzles. By interpolation from the calibration curve, the steam per brake horse-power hour is 42.4 pounds. The steam consumption of the unit based upon useful work is therefore

$$\frac{42.4}{.62} = 68.4 \text{ pounds of steam per air horse power hour.}$$

The dotted line crossing the horse-power characteristics indicates how the power delivered by the turbine varies and also the demand put upon the turbine for delivering various quantities of air into the duct.

In conclusion attention should be called to the fact that the method here presented for pre-determining blower characteristics gives only approximate results. Actual test curves may deviate to some extent from those computed before-hand. If, however, the data obtained from succeeding blower trials is carefully analyzed this method can be made more and more exact, and after some additional experience in manufacturing and operating blowers, the results should be fore-told with almost absolute certainty. The problem that has been carried through in this thesis clearly illustrates how a design can be worked out in a scientific manner so that the final results meet the original requirements. When therefore the machine is installed, it performs its work not wastefully, but with a minimum expense of energy. It should be the ambition

of every engineer to advance the knowledge of his profession so that the problems which may arise, in the future, can be solved with more economical results than have hitherto obtained.